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CONTRACT TRIAL OF THE *OLYMPIA*.

BY ASSISTANT ENGINEER R. E. CARNEY, U. S. NAVY.

The *Olympia* is a steel, twin-screw protected cruiser, of about 5,500 tons displacement, and was built and engined by the Union Iron Works, of San Francisco, Cal., under contract with the Government. The contract price was $1,796,000, with an additional bonus of $50,000 for every quarter-knot in excess of the required average speed of twenty knots per hour for four consecutive hours.

All the material used is of domestic manufacture, the plates for the hull having been furnished by the Carbon Iron Co., of Pittsburgh, Pa., and the shapes by the Midvale Steel Co., of Philadelphia, and the Pacific Rolling Mills, of San Francisco, while the boiler plate came from Carnegie, Phipps & Co., of Pittsburgh, and the shafting from the Bethlehem Iron Co., of Bethlehem, Pa.

She has two masts, the foremast with two and the mainmast with one military top.

The hull is constructed throughout of mild, open-hearth steel. The outside keel plate is $\frac{9}{16}$ inch thick, reduced at the ends to

$\frac{1}{2}$ inch, the inner keel plate and the outside plating $\frac{1}{2}$ inch thick, the former reduced to $\frac{7}{16}$ inch at the ends. The sheer strake is extra heavy, being 1 inch thick, reduced to $\frac{3}{4}$ inch forward and aft, and the inner bottom plating is of $\frac{5}{16}$ inch metal.

There are ninety-one frames, spaced four feet apart within the limits of the double-bottom space, and three feet six inches apart forward of and abaft this space.

On each side of the vertical keel are four longitudinals, of which the two on each side nearest to the vertical keel are water-tight. The fourth one on each side is simply a Z-bar.

Armor.—The protective deck is two inches thick on the flat, and four and three-quarters on the incline, between the limits of the machinery space, the latter dimension being reduced to three inches outside those limits. In the wake of the engine-room hatch, where the tops of the main cylinders require additional vertical protection, a glacis plate four inches thick is worked in, inclined at an angle of thirty degrees.

Battery.—The battery consists of four 8-inch breech-loading rifles mounted in pairs in two barbettes, one just forward of, and the other abaft the central superstructure; ten 5-inch rapid-fire rifles in broadside on the main deck, four 6-pounders on the superstructure deck, under the bridges, and ten 6-pounders on the berth deck. In addition there are to be six Howell torpedo tubes, one at the bow, one at the stern and two on each broadside, fitted to discharge their torpedoes from above water.

Inner Bottom.—The inner bottom extends from frame No. 26 to frame No. 66, inclusive (practically the limits of the machinery space), a distance of one hundred and sixty feet, and, with the frames, longitudinals and vertical keel, forms sixteen double-bottom compartments proper. Further water-tight subdivision below the protective deck is provided for by eleven complete and three partial transverse bulkheads, a complete longitudinal bulkhead extending on the midships line from frame No. 26 to frame No. 66—this being continued forward to frame No. 20 and aft to frame No. 78, as a partial bulkhead—and by the inner bunker bulkheads. Of the three partial transverse bulkheads, two extend as high as the protective deck, but only between bunker bulk-

heads, while the third extends only as high as the after platform or orlop. This subdivision, with the water-tight subdivision above the protective deck, comprises one hundred and thirty-seven compartments, making a total of one hundred and fifty-three water-tight compartments in the ship. Included are the trimming tanks, two forward and two aft, so connected by sluice valves to the regular pumping system that they can be used for giving the desired trim to the vessel. There are forty coal bunkers in the ship, twenty above and twenty below the protective deck, and they, also, form part of the water-tight subdivision. Along each side, at the junction of the protective deck to the hull, and extending longitudinally from frame No. 5 to frame No. 81, is a cofferdam rising vertically to a height of four feet above the load-water line. The forward end, for a distance of eighteen and a half feet, is carried up to the torpedo flat. The compartments of these dams are to be filled with some water-excluding material like cellulose or "Woodite."

Drainage and Pumping.—The main drainage system consists of an 11-inch pipe on each side of and parallel to the vertical keel, and extending from the forward fire room on each side to a well below the forward end of its corresponding engine room, all of the system being within the double bottom. The boiler space comprises four water-tight compartments, and all the surface drainage from the inner skin in the firerooms is taken by a strainer for each compartment—making two for each pipe—and is thence led to the cisterns aft. The drainage of each engine room is led to the main drainage cistern in the forward part of the engine room, or to a well at the after end of same. There is a special drain pipe from the crank pits, fitted with a hand pump, or to discharge into the main cistern through a non-return valve. The bilges throughout are cemented, with midship gutters, and proper depressions for the strainers, which latter, by the way, are fitted with both hand and automatic non-return valves.

The secondary system is of 6-inch piping, half on each side of the midships line, and has communicating valves to connect it to the main system. By means of manifolds, or bilge boxes, and a system of 4-inch piping, any or all of the double-bottom

compartments, or of the lower compartments forward or abaft of them, can be filled with water or emptied at will. The secondary drain extends from the bilge boxes in the forward fire rooms to those at the after end of the engine rooms, the portion in the boiler space being raised above the inner skin. All the suctions to the different compartments have basket strainers, accessible from the inner skin for change or examination. The cisterns to which all this drainage goes are pumped by either of two ways. Ordinarily it is by means of a bilge-box connection, making either the fire and bilge pump or engine-room auxiliary pump available. In case of emergency—as to clear the ship of water in case of a leak—the bilge injection furnishes means of using the centrifugal circulating pumps either to discharge through the condensers or directly overboard. In addition there are four hand pumps—three $7\frac{1}{2}$-inch and one 5-inch—with suctions to the valve boxes, and thus to the secondary drains. Steam-pump suction from the secondary drains is furnished by the fire and bilge and auxiliary engine-room pumps, and by the four auxiliary feed pumps.

SUMMARY OF HULL DATA, AT COMPUTED LOAD DRAUGHT OF 21 FEET 6.6 INCHES.

Length between perpendiculars, feet and inches	340–00
Length over all, feet and inches	344–0¾
Beam, extreme, at L.W.L., feet and inches	53–0⅝
Draught, mean, normal, feet and inches	21–6.6
Area immersed midship section, square feet	1,013
Displacement, tons	5,872.16
Tons, per inch	29 36
Center of buoyancy above base line, feet and inches	12–6
Center of buoyancy aft of midship section, feet and inches	3–4½
Center of gravity above base line, feet	20.517
Transverse metacenter above C. B., feet and inches	10–1¼
Longitudinal metacenter above C. B., feet	353
Distance from center of gravity to transverse metacenter, feet	2.087
Distance from center of gravity to longitudinal metacenter, feet	332.53
Coefficient of displacement	0.520
Coefficient of midship section	0.886
Coefficient of L.W.L.	0.744
Cylindrical coefficient	0.587
Area of L.W.L. plane, square feet	13,421.4
Wetted surface, square feet	24,864.0

Main Engines.—There are two vertical, inverted, direct-acting, three-cylinder, triple expansion engines, side by side, with a midships water-tight bulkhead between them, the engine rooms being connected at the level of the middle platform by two water-tight doors. The cylinders are of the best quality of cast iron, and all are fitted with linings of hard, close-grained iron, the annular spaces around the I.P. and L.P. linings forming steam jackets for those cylinders. The linings are accurately bored to the following diameters: H.P., 42 inches; I.P., 59 inches, and L.P., 92 inches, and counterbored at both ends, leaving a working length of barrel of 3 feet 8¼ inches, for a 42-inch stroke. The valve gear is of the Stephenson type, with double-bar links working the valve directly, the cut-off for full gear taking place at about .7 of the stroke. Within small limits the travel of each valve is adjustable by means of four slots in the reversing arms. The valves are of the piston type, those for the H.P. being balanced by making the upper half of the valve ⅛ inch greater in diameter than the lower half, while the intermediate and low-pressure valves are fitted with balance pistons. The reversing gear consists of a steam cylinder and a hydraulic controlling cylinder, the piston rods of both fitted to the same crosshead, and the ends of the hydraulic cylinder connected with a by-pass worked from the stem of the steam starting valve. These valves receive their primary motion from a hand lever at the starting platform, and are stopped by a system of levers receiving motion from a pin on the reversing arm, the parts being so adjusted in this system that the reversing engine follows the motion of the hand lever, and when stopped is firmly held by the controlling cylinder. There is also a small hydraulic pump for reversing by hand.

Each engine has a 15-inch gate stop valve, and a gridiron throttle valve, the latter worked from the lower platform by a hand wheel, worm and gear.

The main pistons are of cast steel and dished, and each has two packing rings 1¼ inches wide and ⅜ inch thick, set out by steel U-springs. The piston rods, connecting rods and crossheads are of forged steel, the latter being fitted with composition

slippers faced with hammered white metal. The engine framing consists of one inverted Y-frame at the back of each cylinder, and a pair of forged-steel columns in front. The Y-frames are built up, having wrought-iron side pieces secured to a cast-iron body piece which carries the crosshead slide, and to which the backing slides are bolted. Both Y-frames and columns are secured at the bottom to manganese bronze bed plates, the latter in turn being held down by forged-steel bolts. The crosshead guides are cored out for circulating cooling water. The eccentrics are in two parts, of cast steel, keyed on to the shaft, the go-aheads with adjustable side keys. The eccentric straps are of composition, while the rods, with the links, link blocks, suspension rods and valve-stem crossheads, are of forged steel. The exhaust system between valve chests is external, and consists of copper piping. All important valves in connection with the steam system, including the safety valves, are fitted to be operated either from the berth deck or from below.

ENGINE DATA.

Number of cylinders, each engine	3
Diameter of cylinders, H.P., inches	42
Diameter of cylinders, I.P., inches	59
Diameter of cylinders, L.P., inches	92
Ratio of piston areas	1.0 : 1.973 : 4.798
Diameter of piston rods, inches	8¼
Stroke, inches	42
Length of connecting rod between centers, inches	84
Volume swept by L.P. piston, in cubic feet	161 574
Diameter of H.P. valve, one for each engine, inches	Top 18½; Bottom 17⅞
Diameter of I.P. valve, two for each engine, inches	23
Diameter of L.P. valve, four for each engine, inches	21

Shafting.—The shafting is steel throughout, and all shafts are lightened by axial holes. There are three crank shafts for each main engine, interchangeable and reversible, and so joined as to bring the cranks 120° apart, in the following sequence: H.P., I.P., L.P. The length of each section is equal to the distance between the axes of the piston rods, 9 feet 7½ inches. The crank pins are 21½ inches long and 17 inches diameter. The

crank webs are 10¼ inches thick, and taper from 18½ to 17½ inches in width. The axial holes in crank pins and journals are respectively 8½ and 7½ inches. The coupling flanges at the ends of these sections are 2 feet 7 inches in diameter, and 3⅝ inches thick, each drilled for six forged steel bolts, 3½ inches diameter, with wrought-iron nuts and split pins. At each end of each section is a raised eccentric seating 9 inches long and 17 inches in diameter.

Each thrust shaft is 19 feet 5¼ inches long, and has twelve collars, two inches wide, and separated from the adjacent one by 3½-inch spaces. The collars are 20½ inches in diameter, and the shaft 15¾ inches. Just abaft the forward flange is a raised eccentric seating for the after L.P. eccentrics, similar to those on the crank shafts. The thrust shafts also have bearing journals forward and abaft the collars to take the weight off of the latter, and are lightened by 7½-inch axial holes.

The line shafts are each 35 feet long, 15¾ inches diameter, lightened by a 7½-inch axial hole, and have a socket at the after end into which is fitted the tapered forward end of the propeller shaft. An independent disc or flange is screwed on this taper, and serves as a coupling flange for the propeller shaft, which is also secured by feather keys.

The propeller shafts are each 39 feet long, including the taper at inboard end, which extends two feet into the socket of the line shaft; they are 16 inches in diameter, with a 7½-inch axial hole. The shafts are cased with a composition sleeve ⅝ inch thick, except where it forms a journal, when it is increased to 1 inch.

The main bearings are fitted only with bottom brasses, the cap itself serving as the other brass, both it and the bottom brass being lined with white metal. The thrust bearing is of cast iron, the collars having white metal facings. The cap is similarly fitted. The body of the bearing has a trough, with stuffing boxes around the shaft to be filled with oil, which is supplied to the thrust facings through holes between the collars. A cooling coil connected with the engine-room water service keeps the liquid in the trough cool. The holes for holding down bolts are elliptical to allow for adjustment. At the forward and after ends

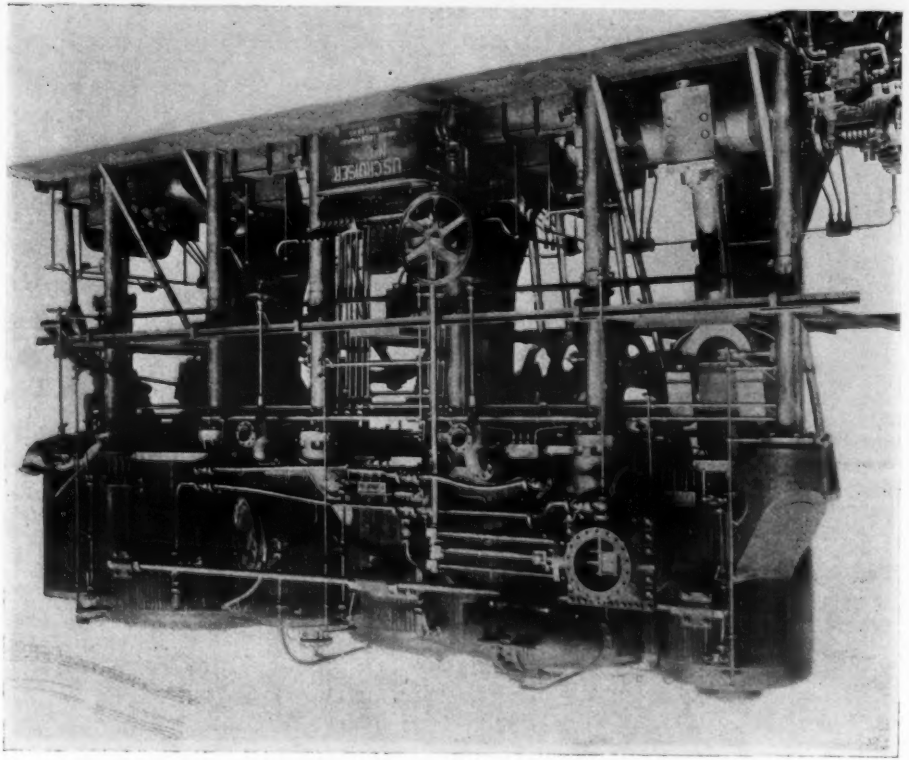
of the trough are ordinary bearings lined with white metal, and capable of vertical adjustment to take the weight of the shaft from the thrust collars.

In each stern tube are composition bearings about 3 feet long each, and fitted with lignum vitæ on end of grain. The strut bearings are similarly fitted with composition bushings and lignum vitæ.

Screw Propellers.—There are two three-bladed screw propellers, the starboard one right, and the port one left handed. They are of manganese bronze, 14 feet 9 inches in diameter, and the pitch is adjustable between 18 feet 6 inches and 19 feet 6 inches. During the trial it was 19 feet mean. The disc area of each is 170.9 square feet, the helicoidal area 68 square feet, and the ratio of the diameter to the mean pitch 1 to 1.288. The hubs are 4 feet diameter. During the trial the average submersion of the tip of the upper blade was 83 inches.

Main Condensers.—There are two main condensers of cast naval brass, 6 feet 6 inches internal diameter, 11 feet 10 inches long between tube sheets. In each condenser there are 4,904 seamless drawn brass tubes, making a cooling surface of 9,495 square feet for each main engine. The tubes are $\frac{5}{8}$ inch outside diameter, No. 20 B.W.G., and the circulating water passes through them. The ratio of cooling surface of main condensers to total heating surface is about 1 to 1.498.

Air Pumps.—For each main condenser there is a triple air pump, consisting of three vertical single-acting pumps, driven by a like number of simple inverted engines, one engine cylinder over each pump cylinder. In each case there is a steel piston keyed into the upper side of a bronze crosshead, into the lower side of which is keyed a pump rod of hammered phosphor bronze. The crosshead slippers are fitted with white metal bearing surfaces. The crosshead pins are of forged steel, as are also the crank shafts, valve rods, and connecting rods. Heavy cast-iron balance wheels at the ends of the crank shafts secure uniformity of rotation. The cranks are 120° apart, and the common stroke of pumps and engines is 20 inches. The steam cylinders are $5\frac{1}{2}$ inches in diameter, fitted with rocking slide valves, and



steam is supplied from the auxiliary steam system. The exhaust is either by a special opening into the main condensers, or into the auxiliary exhaust system. The air-pump barrels are of composition, 20 inches in diameter, with composition pistons, valve seats and guards, while the valves and springs are of phosphor bronze. For each pump there are ten foot valves, eight bucket valves and eight head valves, the latter at the top of the pump chamber. The seats for the foot valves are inclined. For each set of pumps there is a common suction box, just back of and below the foot valves. This has a stop valve in each pump suction, so that any pump can be shut off at will. Behind the head valves is the delivery chamber, and from this the condensed feed water is drawn by a small "tank pump" and forced to the feed tanks in the fire rooms.

Circulating Pumps.—Circulating water for each condenser is supplied by two centrifugal pumps, fitted to draw either from the sea or from the bilge of its engine room, and to deliver into the condenser, or directly overboard by means of a pipe connecting the condenser supply with the outboard delivery. There is a safety locking device fitted to main and bilge injection valves so that one cannot be opened unless the other is tightly closed. The runners of the pumps are 28 inches in diameter, with 12-inch suction and delivery pipes, and each is driven by a compound engine of 6-inch stroke, the cylinders being $5\frac{1}{2}$ and 11 inches diameter respectively. Each pump is capable of discharging 6,750 gallons of water per minute from the bilge.

Boilers.—There are four double-ended and two single-ended steel boilers with horizontal return fire tubes, the former with eight and the latter with four corrugated steel furnaces each. The boilers are placed in four water-tight compartments, one double-ended boiler in each forward, and one double and one single-ended boiler in each after compartment. The total grate area is 824 square feet, and the total heating surface 28,298.64 square feet. All the top heads are curved to save staying. The longitudinal seams are butted, strapped inside and out with 1 inch straps, and treble riveted. The circumferential seams between courses are lapped, the middle course being smallest, and

treble rivetted, and the joints of the heads with the shells are double riveted. The joints at the furnace mouths and those in the combustion chambers are single riveted. The plates are of open-hearth steel, and the rivets of Clapp-Griffiths steel. The grate bars are of wrought iron, of the shaking pattern, and can be removed or placed without hauling fires. Each single-ended boiler has six, and each double-ended boiler twelve circulating tubes of oval cross-section, in addition to which the internal feed pipes and circulating plates provide for proper circulation of water in the boilers. Hydrokineters of the Weir type are also fitted. The forward smoke pipe is a flat-sided oval, with its greatest diameter in a fore-and-aft direction, and it receives the uptakes of the forward pair of double-ended boilers. The after smoke pipe is circular in cross section and receives the uptakes of the remaining boilers. The tops of the smoke pipes are 75 feet above the lowest grates. The tubes are of steel, lap-welded and drawn, the plain tubes of No. 12, and the stay tubes of No. 6 B.W.G., all $2\frac{1}{4}$ inches outside diameter. The stay tubes are screwed at both ends, and protected at the fire ends by cast-iron ferrules. The plain tubes are expanded and beaded over at both ends.

BOILERS.

Steam pressure, pounds	160
Length, double-ended, feet and inches	21–3
single-ended, feet and inches	10–11½
Diameter, feet and inches	15–3
Thickness of shell plates, inches	$1\frac{19}{64}$
top heads, inches	$1\frac{19}{64}$
tube sheets, inch	$\frac{7}{8}$
lower front heads, inch	$\frac{3}{4}$
back heads, single-ended, inch	$\frac{3}{4}$
butt straps, inch	1
combustion chamber sheets, inch	$\frac{1}{2}$
Rivets, diameter for circumferential seams, inch	$1\frac{5}{16}$
longitudinal seams, inch	$1\frac{1}{4}$
heads, inch	$1\frac{3}{16}$
combustion chamber sheets, inch	$\frac{3}{4}$
Braces above tubes, diameter, inches	$2\frac{3}{8}$
to back tube sheet, diameter, inches	2 and $2\frac{1}{8}$

Braces above lower man-holes, diameter, inches	2
diagonal in bottom water space, inches	1⅝
Screw stays, diameter, inches	1 1/16
pitch, inches	7
Furnaces, inside diameter, inches	39
outside diameter, inches	43
thickness, inch	17/32
number in each double-ended	8
in each single-ended	4
total	40
Grates, length, feet and inches	6–4
Combustion chambers, number in each double-ended	4
number in each single-ended	2
Tubes, plain, number in each double-ended	796
number in each single-ended	398
outside diameter, inches	2¼
stay, number in each double-ended	268
number in each single-ended	134
outside diameter, inches	2¼
length between tube sheets, feet and inches	7–7⅞
spacing, horizontally, inches	3½
vertically, inches	3¼
total number in each double-ended	1,064
total number in each single-ended	532
Heating surface, plate, each double-ended, square feet	776.40
plate, each single-ended, square feet	388.20
tube, each double-ended, square feet	4,883.33
tube, each single-ended, square feet	2,441.67
total, each double-ended, square feet	5,659.73
total, each single-ended, square feet	2,829.87
total, all boilers, square feet	28,298.64
Grate surface, each double-ended, square feet	164.8
each single-ended, square feet	82.4
total, square feet	824.0
Ratio heating to grate surface	34.3
Steam space, each double-ended, water 6 inches above tubes, cubic feet	990.41
each single-ended, water 6 inches above tubes, cubic feet	564.90
Water surface, each double-ended, same conditions, square feet	287.21
each single ended, same conditions, square feet	138.02
Safety valves, number each double-ended	3
number each single-ended	2
diameter, inches	5
load on, pounds per square inch	168
Stop valves, double-ended boilers, diameter, inches	10
single-ended boilers, diameter, inches	7

Forced Draft.—Air pressure is maintained in the fire rooms by eight blowers—two for each air-tight compartment in the boiler space—the engines being of the compound vertical type, with cranks 180° apart, cylinders 5 and 8 inches diameter respectively, by 6 inches stroke. The fans are 66 inches in diameter, each composed of eight curved vanes, and deliver directly into the fire rooms, except that light iron baffle plates are fitted to prevent the blowing about of coal dust.

Steam Piping.—The main steam pipes, from the throttles back to the main stop valves are of No. 00 B.W.G. copper, 15 inches in diameter. Extending forward of this through the fire rooms the piping is 12 inches in diameter, with a 10-inch branch to each main boiler stop valve, these branches being No. 0 B.W.G. Connecting the two main separators is a 12-inch copper pipe. All the main steam system of piping is strengthened by steel bands 2 inches wide, ½ inch thick, and spaced eight inches from center to center. An auxiliary steam pipe, 6 inches in diameter, extends throughout the engine and boiler spaces, with connections to all auxiliary machinery therein, and additional branches to the windlass and ship's ventilating engines, to the engineer's work shop, and to the dynamo and turret engines. This pipe connects with the auxiliary stop valves of all boilers, and with the main steam pipe in each engine room abaft the separator. Stop valves are also fitted wherever the system cuts a transverse water-tight bulkhead. All downward branches are fitted with stop valves close to the auxiliary steam pipe to prevent accumulation of condensed water while the branch is idle, all condensation in the auxiliary steam pipe proper draining either back to the separator or into traps.

Feed Pumps.—There are four main and four auxiliary feed pumps in the fire rooms containing the boiler check valves, *i. e.*, in the four central fire rooms. The arrangement of the fire-room pumps is as follows: In each of fire rooms "E" and "F" are two vertical direct-acting pumps of 330 gallons per minute capacity, the steam cylinder being 12 inches diameter, the water cylinder 8 inches, and the stroke 16 inches. Of each pair, one pump is outboard and the other inboard. In each of fire rooms

"C" and "D" are two similar pumps to those above described, except that the stroke is reduced to 12 inches, and the capacity to 220 gallons per minute. As before, one pump of each pair is outboard and one inboard, and in all cases the outboard pumps supply the *main* feed, and the inboard ones the auxiliary. The main feed pumps draw only from the feed tank and deliver into the main feed pipe, while the auxiliary pumps take suction from either sea, feed tank, bilge or boilers, and deliver into the auxiliary feed pipe, fire main, or overboard through the outboard delivery in its own compartment.

Feed Tanks and Filters.—The feed tanks are built in between the forward and after boiler compartments, and each has a capacity of 1,700 gallons. A portion of each tank is rigged as a filter, and during the trial coarse gunny sacks loosely stuffed with "Excelsior" were used in the filter spaces with excellent results. Water is drawn from the hot wells by the tank pumps, and forced to the feed tanks, the auxiliary air pumps discharging into the same. The tanks are fitted with glass gauges and drain cocks, and each has an overflow to the bilge so arranged that any escape of excess feed water is noticeable. In addition each tank has a suction pipe for the main, and one for the auxiliary feed pumps on its side of the ship, as well as for the corresponding auxiliary engine-room pump.

Distilling Plant.—This consists of two evaporators and two distillers, with a combined capacity of 8,100 gallons of potable water per day. For both evaporators and distillers there is one combined air and circulating pump, which provides cooling water for the distiller, and by connection to the end of each coil evaporation is carried on at or below the atmospheric pressure. The circulating water then passes to the salt-water tank on the upper deck, while the fresh water is pumped into the tanks below, to the supply tank on deck, or to the feed tanks. There is also a hand-pump connection to the firemen's wash-room tank. The evaporators are of the improved Baird type, horizontal, the shells being of plate steel, lapped and riveted, the heads and flanges of cast iron and the coils of copper, tinned inside and out. The distillers are of sheet brass, with tinned copper coils, fitted with filters, and with the parts accessible for cleaning and repairs.

Auxiliary Condensers.—There is an auxiliary condenser in each engine room, connected by the auxiliary exhaust system with all the auxiliary machinery in the ship, each condenser capable of caring for one half. The shells, heads and tube sheets are of composition, while the tubes are of the same size and kind as those fitted in the main condensers, and are similarly packed, *i. e.*, with cotton cord and screwed glands. The condensing water is passed through the tubes, and the total cooling surface for each is 455 square feet. Under each condenser is a combined air and circulating pump of the horizontal, double-acting type, both pumps driven by an 8-inch steam cylinder. The air-pump barrels are 11 inches, and those of the circulating pumps 9 inches in diameter, with a common stroke of 10 inches.

Additional Engine-Room Pumps.—At the after end of each engine room is a pair of pumps, duplicates, fitted with the Dow valve gear and governors. (In fact, all pumps in the ship have the latter appliance, and all but the main air and circulating pumps have the valve gear.) These four pumps are of the vertical, direct-acting type, of 500 gallons per minute capacity each at 60 pounds pressure of steam, with 10-inch steam and 8-inch water cylinders, and an 18-inch stroke. All are fitted to take suction from the hot wells, condenser, sea, bilge of compartment next abaft, secondary drain pipe and drainage cistern, and to discharge into feed tanks, fire main or overboard. One of each pair is designated a "fire and bilge pump," and the other an "auxiliary engine-room pump." In each engine room is a small, vertical, direct-acting pump of 150 gallons per minute capacity, for supplying the water service, and another, the "tank pump," to draw from the hot well and force the condensed water to the feed tanks.

Steering Gear.—The ship is fitted with hydraulic steering gear consisting of a pair of rams working a yoke on the rudder head. The actuating pressure is 500 pounds to the square inch, and is maintained by a pair of hydraulic pumps at the after end of each engine room. There is no accumulator, but each pair of pumps has a governing throttle, loaded in proportion to the desired pressure, so that the latter is always maintained.

Ice Machine.—Forward on the main deck is a 1½-ton ice machine, of the Allen dense-air type, and just abaft it on the same deck is a large refrigerating room.

Dynamos.—There are four dynamos, each of 400 amperes and 80 volts, and driven by compound engines of the vertical, inverted, direct-acting type, with cylinders 7½ and 13½ inches in diameter, and a 6-inch stroke.

Turret Engines.—The training power for each pair of 8-inch guns consist of a two-cylinder, simple engine, with cylinders 8 inches in diameter and a 6-inch stroke. The engine framing is bolted to a vertical bulkhead below the protective deck, and the crank shaft, by bevel gearing, actuates two worm wheels which gear into opposite sides of a large spur wheel on the foot of the ammunition tube, the latter forming the revolving spindle of the double gun carriage. The limits of train depend on the superstructure of the ship, and automatic stops in the controlling gear of the engines prevent any train in excess of these limits.

WEIGHTS.

Propelling machinery, including water, tons			1,215.23
Reciprocating parts:	*Starboard.*		*Port.*
H.P. piston, pounds	1,682		1,683
I.P. piston, pounds	2,821		2,787
L.P. piston, pounds	5,603		5,540
Piston rods, pounds		4,623	
Connecting rods, pounds		17,874	
Crossheads, pounds		5,809	

THE TRIAL.

After several failures to complete a trial, due to weather and other ungovernable circumstances, the official trial took place in Santa Barbara Channel, Cal., on December 15, 1893. It consisted of a run in each direction over a course 42.9843 nautical miles in length, the mean speed corrected for tide being 21.686 knots.

The previous attempts all stand to the credit of the ship, as altogether the engines and boilers were worked up to the maximum conditions on five different occasions, and after the official trial none but routine cleaning and overhauling was necessary.

In connection with the latter it must be stated that the engineer's department is excellently fitted with appliances for the removal of parts for examination or overhauling. Everything worked excellently during the trial, the boilers furnishing plenty of steam, with very slight variation in pressure. There was comparatively little vibration of the main engines, and this is attributed to the athwartships tie rods between the engines, by means of which *both* bed plates serve for each engine, instead of being separate. As a precautionary measure the water service about the crank shafts was in use. Both auxiliary condensers were in use, receiving the exhaust of all the auxiliaries. Indicator diagrams were taken every half hour from each main cylinder, and every hour from the main air and circulating pumps. The weather conditions were favorable, and the ship maintained a corrected average speed of 21.686 knots.

DATA.

Mean draught, forward, feet and inches	19–3½
Mean draught, aft, feet and inches	22–2
Draught, mean, feet and inches	20–8¾
Displacement, tons	5,586
Tons, per inch at mean draught	29.25
Center of buoyancy above base, feet and inches	12–1⅛
Transverse metacenter above C.B., feet and inches	10–6⅝
Longitudinal metacenter above C.B., feet and inches	366–0⅝
Block coefficient of fineness	.517
Average speed, knots	21.686
$Speed^3 \times$ area immersed midship section $\div$ I.H.P.	579.31
$Speed^3 \times displacement^{\frac{2}{3}} \div$ I.H.P.	187.67

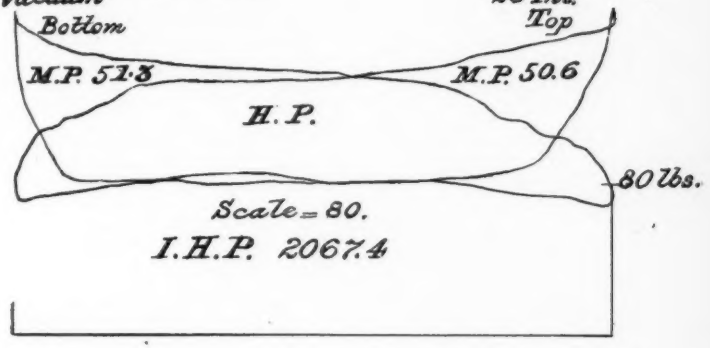
SYNOPSIS OF STEAM LOG.

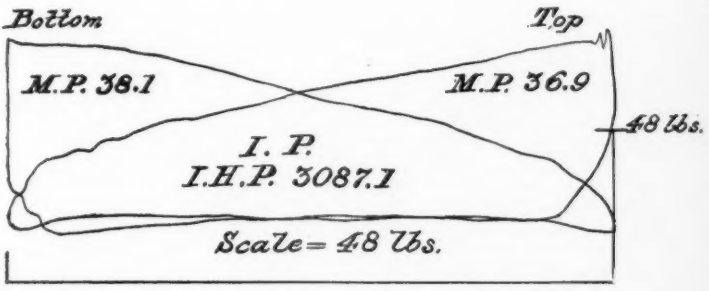
	Starboard.	*Port.*
Revolutions of main engines per minute	139.98	138.53
Revolutions of main engines, mean	139 25	
Piston speed, feet per minute	979.86	969 71
Steam at boilers, (gauge)	166.53	
Steam at engines, (gauge)	163 75	164.83
Steam at first receiver (absolute)	92.82	95.6
Steam at second receiver (absolute)	31.11	33 53
Vacuum in condenser, inches	24.94	25 59
Opening of throttle	Wide.	
Steam cut-off in decimals of stroke from beginning, H.P.	.770	.776
I.P.	.731	.735
L.P.	.735	.735

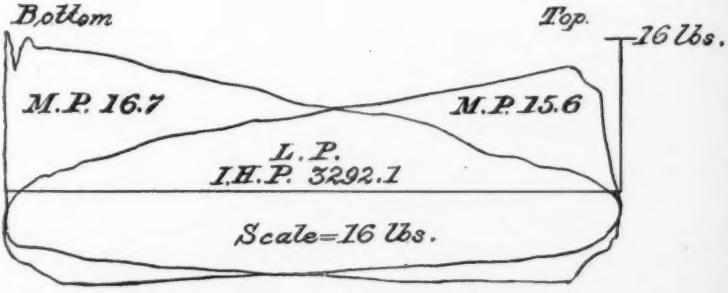
U.S.S. Olympia.

Star'd. Eng.

Dec. 15th 1893 — 1.09 P.M.
Steam Pressure at Eng. — 162 lbs.
Revolutions per Min. — 141.
Vacuum — 25 Ins.



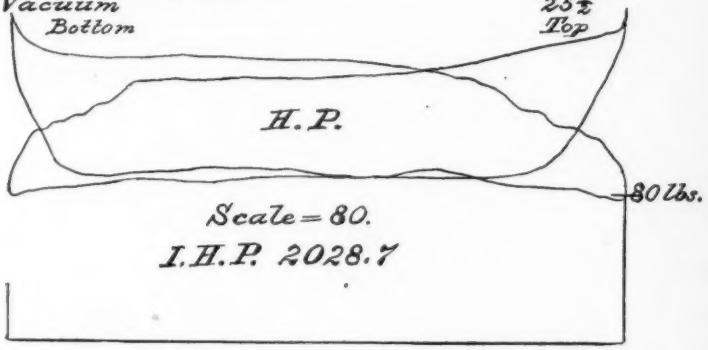


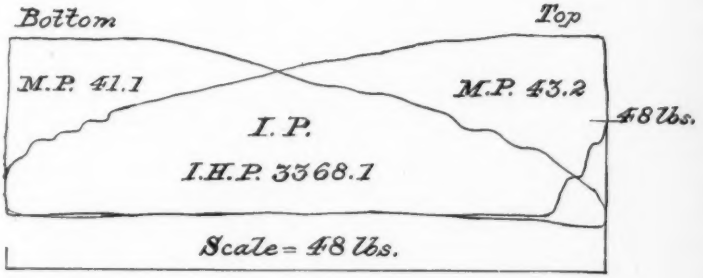


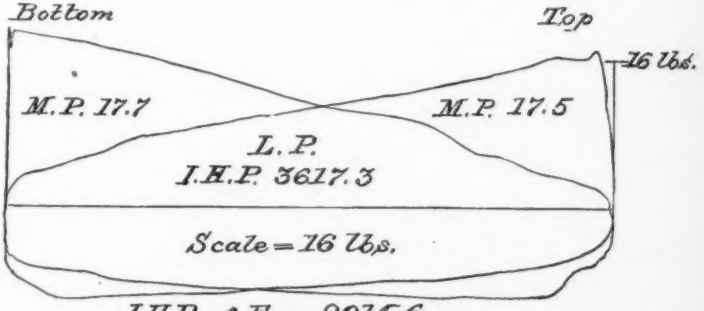
U.S.S. Olympia.

Port. Eng..

Dec. 15th 1893. 1.11 P.M.
Steam Pressure at Eng. 165 lbs.
Revolutions per Min. 138.5
Vacuum 25"







I.H.P. of Eng. 9014.6

	Starboard.	*Port.*
Revolutions of air pumps	162.	124.7
Revolutions of circulating pumps	344.	369.1
Tempertures in degrees Fahrenheit, injection	60.	60.
discharge	102.	105.5
hotwell	121.5	
feed tank	130.	106.
Revolutions of blowers	470.	475.
Air pressure in fire rooms, average, inches	2.04	
H.P. cylinder, mean pressure	49.63	47.65
I.H.P.	2,001.	1,903.2
I.P. cylinder, mean pressure	38.51	40.05
I.H. P.	3,097.3	3,185.6
L.P. cylinder, mean pressure	16.27	17.8
I.H.P.	3,198.5	3,463.4
Equivalent pressure on L.P. piston equal to aggregate M.E.P.	42.45	44.18
Collective I.H.P. of each main engine	8,297.6	8,552.2
Collective I.H.P. of both main engines	16,849.8	
I.H.P. of air pumps, separate	18.22	35.78
I.H.P. of circulating pumps, in pairs	66.50	65.20
Collective I.H P. of air and circulating pumps	84 72	100.98
I.H.P. of air pumps combined	54.00	
I.H.P. of circulating pumps combined	131.7	
I.H.P. of main feed pumps (four)	44.7	
I.H.P. of combined auxiliary air and circulating pumps	31.28	
I.H.P. of eight blower engines	130.56	
I.H.P. of water-service pump	7.14	
I H.P. of fire and bilge pumps	10.66	
I.H.P. of engine-room auxiliary pumps	5.53	
I.H.P. of hot well (tank) pumps	17.96	
I.H.P. of dynamo engines, two	29.75	
Collective I.H.P. of all auxiliaries in use except main air and circulating pumps	277.58	
Collective I.H.P. of each main engine, with its air and circulating pumps	8,382.32	8,653.18
Collective I.H P. of both main engines, with main air and circulating pumps	17,035 50	
Collective I.H.P. both main engines, with all auxiliaries	17,313.08	
Kind and quality of coal	Best Harris Steam Navigation.	
Pounds per hour	37,937.5	
Pounds per hour per I.H.P., collective of all machinery in operation	2.19	
Pounds per hour per I.H.P., collective of main engines, main air and circulating pumps	2.22	
Pounds per hour per square foot of heating surface	1.38	

	Starboard.	*Port.*
Pounds per hour per square foot of grate surface		44.7
Square feet of cooling surface per I.H.P		1.152
Square feet of heating surface per I.H.P		1.635
I.H.P. per square foot of grate		21.011
I.H.P. per square foot of screw area	122.02	125.77
Helicoidal area of both screws to immersed midships section		.1402
Ratio of disc area of both screws to immersed midships section		.3524
Indicated thrust, pounds (I.H.P. main engines)	102,954.86	107,224.59
Indicated thrust per square foot of developed area of propeller, pounds	1,514.04	1,576.83
Indicated thrust per square inch of surface of thrust-bearing pounds	63.44	66.07
Weight of propelling machinery (water included), tons		1,215.23
Weight of propelling machinery per I.H.P., pounds		157.23
I.H.P. per ton of machinery (maximum I.H.P.)		14.247
Slip of propellers, per cent	17.3	16.44

The data above given in relation to coal consumption were arrived at by the use of bagged coal for the eight furnaces in the after fire rooms, the bags containing 100 pounds each, and the rate thus reached was assumed as the mean rate for all the furnaces.

Reference to the tabulated performances will show a peculiar condition of affairs, in that the port engine developed 254.6 H.P. more than the starboard engine while averaging 1.45 revolutions less per minute. But a comparison of the equivalent mean pressures to develope the total horse-power in the L.P. cylinders shows it to be 42.45 pounds per square inch on the starboard side, and 44.18 pounds on the port side, from which the inference would seem to be that the port propeller is coarser in pitch than the starboard one. The very high equivalent pressure given above is worthy of note, comparing most favorably with that shown for recent ships.

[It is proper to state that the indicators used on this trial were not tested in the manner usual in such cases, or in trials for indicated horse-power, but by comparison with a standard spring pressure gauge which had been tested by the mercury column at the Mare Island Navy Yard. The method usually followed is described by Assistant Engineer Conant, U. S. Navy, on page 322 of Volume III, of the Journal.—ED.]

THE PROPULSIVE POWER OF SCREW SHIPS.

By James N. Warrington, Esq., Associate.

Since the investigations of the elder Froude, friction and wave making have been fully recognized as the leading elements in the resistance of ships. The resistance due to eddies is known to be quite inconsiderable in modern ships, while the variations in resistance which accompany change in trim, bodily rise or subsidence of the vessel, and wave interference, all occur at such speeds as produce waves of great magnitude relatively to the displacement. Within the limit of speed at which these phenomena appear, friction and wave making may be regarded as supplying substantially the entire resistance, and it is here essayed to formulate the propulsive power absorbed by these two elements of resistance with such a degree of approximation as may be attained by attributing the properties of a trochoidal wave to the waves of the ship.

FRICTION.

The horse-power of the frictional resistance may be expressed by the formula

$$P_f = \frac{SV^n}{C_1}, \qquad (1)$$

in which S represents the area of the wetted surface, V the speed in knots, and C_1 a constant to be derived from the coefficient of friction. For a clean painted surface exceeding fifty feet in length Founde's experiments show a resistance of 0.25 lb. per square foot at a speed of ten feet per second, and 1.83 as the power of the speed to which the resistance is proportional. From these quantities are deduced the values $n = 2.83$ and $C_1 = 32{,}566$, the speed being expressed in knots of 6,080 feet and the wetted surface in square feet.

WAVE MAKING.

Considering the waves generated by the ship in motion, it may be observed that similar ships produce similar waves, and that for a given ship the ratio of the magnitude of any series of waves to the magnitude of any other series, as for example of the bow waves to the stern waves, will be constant at all speeds so long as their generation is free from mutual interference. Moreover the waves of any series will retain their similarity at all speeds, for the orbital velocity as well as the speed of advance of the wave form must both be proportional to the speed of the ship.

The aggregate energy of all the waves of the ship may therefore be conceived to be embodied in a single series of representative waves having a speed of advance equal to the speed of the ship.

As the particles of water in the longitudinal stream lines approach the ship they meet the component stream lines diverging from the apex of the bow wave and follow the resultant stream lines around the ship. The velocity in the diverging stream lines must clearly be sufficient to impart the necessary transverse displacement during the time occupied in the passage from the bow to the midship section. The diverging stream-line volocity will then be proporional to $\frac{VA^{\frac{1}{2}}}{L}$, in which V is the speed of the ship, A the area of midship section, and L the length of combined entrance and run. But this diverging stream-line velocity is closely proportional to, if not identical with, the orbital velocity of the bow waves. Hence, these orbital velocities also will be proportional to $\frac{VA^{\frac{1}{2}}}{L}$, and since the waves of each series grow in like proportion with increase of speed, the orbital velocity of the representative wave also will be proportional to the same ratio.

Assuming now that the waves of the ship possess the properties of trochoidal waves, it follows that the height of the representative wave will be proportional to $\frac{lu}{V}$, in which l represents the wave length, u the orbital velocity, and V the speed of ad-

vance. Furthermore, the breadth of the wave will be proportional to its height, for as the height increases the orbital velocity increases throughout the entire breadth of the wave, and imparts its motion to more remote particles previously at rest.

The horse-power of a series of trochoidal waves, having a length l, a height h, and a breadth h, is thus expressed by Mr. Albert W. Stahl, U. S. Navy:*

$$0.0329\, h^3\, l^{\frac{1}{2}} \left(1 - 4.935 \frac{h^2}{l^2}\right). \qquad (2)$$

The ratio $\frac{h^2}{l^2}$ is constant for similar ships and for all speeds of a given ship, and is proportional to $\frac{A}{L^2}$. Since with such a range in this latter ratio as may be found between a torpedo boat and an armored cruiser, the quantity in parenthesis would vary by a fraction of one per cent., it may reasonably be neglected. The wave horse-power then becomes proportional to $h^3\, l^{\frac{1}{2}}$. But $l^{\frac{1}{2}}$ is proportional to V, and h to $\frac{V^2 A^{\frac{1}{4}}}{L}$; hence the horse-power absorbed in wave making may be expressed by the formula

$$P_w = \frac{A^{\frac{3}{4}} V^7}{C_2 L^3}, \qquad (3)$$

in which C_2 is a constant to be derived from the performance of a model ship.

The imaginary representative wave is conceived as having a speed of advance equal to the speed of the ship, and in this respect is plainly in harmony with the transverse series. The speed of advance of the diverging waves is dependent not only upon the speed of the ship but also upon the angle of divergence of their crest lines, and this angle of divergence is dependent upon the bluntness of the ship, or in other words upon the ratio $\frac{A^{\frac{1}{4}}}{L}$, so long as the mode of distribution of the displacement remains the same. The length of the diverging waves will then vary with this ratio, and the effect will be that in the blunter ship the diverging series will absorb a greater amount of energy rela-

* Transactions American Society of Mechanical Engineers, Vol. XIII, p. 454.

tively to the transverse series than in the ships of finer form. This relative increase of energy in the diverging series must, however, be accompanied by a corresponding diminution of the energy in the transverse series, for the total energy in both series is derived from the diverging component stream lines, the velocity in which is proportional to $\frac{VA^{\frac{1}{6}}}{L}$; and this total energy thus derived will be independent of the relative importance of the two waves. The condition necessary to the preceding statement is that the model ship, whence the constant is derived, and the projected ship shall be similar in cross section and identical in mode of distribution of the displacement. When this condition is fulfilled, therefore, the representative wave becomes a true measure of all the waves subject to the approximation involved in its trochoidal nature, and in the modification of expression (2).

Concerning the limit of speed in the application of the formula, it should be observed that each wave-making feature is assumed to generate its appropriate wave without interference; hence the beginning of each interference marks the limit of applicability.

At corresponding speeds the power absorbed in wave making becomes proportional to the square root of the seventh power of the scale of comparison, as should be expected.

THE WAKE CURRENT.

The sum of P_f and P_w is the effective horse-power required to tow the ship. When propulsion is effected by a screw propeller at the stern, other varying elements are introduced. The following current, by delivering momentum to the propeller, restores a portion of the energy already expended in overcoming the frictional and wave-making resistances.

Let V represent the speed of the ship, v the mean speed of the following current, and d the diameter of the propeller. Then the energy imparted to the propeller per unit of time by the following current will be proportional to

$$d^2 v V(V-v); \qquad (4)$$

but assuming a constant percentage of real slip, d^2 is proportional to

$$\frac{P_f + P_w}{V^2(V-v)},$$

which being substituted in (4) reduces the increment of energy to the form

$$\frac{v(P_f + P_w)}{V}. \tag{5}$$

The following current has three components of independent origin, viz: the frictional wake, the orbital velocity of the stern wave, and the orbital velocity of the recurring transverse bow wave. It has been shown that when the waves of the bow series interfere with the formation of the stern wave the limit to the application of this analysis has been reached. Therefore the effect of the bow wave need not be considered.

The speed of the frictional wake will here be regarded as proportional to the speed of the ship; hence the energy derived from it, being a constant proportion of the total power, need not be considered here.

Still attributing the properties of a trochoidal wave to the waves of the ship, the orbital velocity of the stern wave at the center of the propeller will be proportional to

$$\frac{VA^{\frac{1}{2}}}{L_a e^{\frac{2\pi h}{\varepsilon V^2}}}, \qquad e^{\frac{2\pi h}{\varepsilon V^2}} \tag{6}$$

in which L_a is the length of the after body, e the modulus of the hyperbolic system of logarithms, h the depth of immersion of the axis of the propeller, and ε a constant such that εV^2 equals the wave length.

Substituting (6) for v in expression (5) the increment of energy becomes

$$(P_f + P_w)\frac{A^{\frac{1}{2}}}{C_3 L_a e^{\frac{2\pi h}{\varepsilon V^2}}}. \qquad e^{\frac{2\pi h}{\varepsilon V^2}} \tag{7}$$

The value of the constant C_3 as derived from the results of the experiments of Mr. George A. Calvert* is $C_3 = 1.66$; but this

* Paper read at thirty-fourth session of the Institute of Naval Architects, July 13, 1893. Published also in Journal American Society of Naval Engineers, Vol. V, p. 869.

value should be regarded as provisional, since C_3 like C_2 depends upon the mode of distribution of the displacement.

SUMMARY.

Let P_t represent the total effective horse power required at the propeller. Then

$$P_t = (P_f + P_w)\left[1 - \frac{A^{\frac{1}{2}}}{C_3 L_a \varepsilon^{\frac{2\pi h}{\sqrt{A}}}}\right], \qquad (8)$$

in which $P_f = \frac{SV^{2.83}}{C_1}$, $P_w = \frac{A^{\frac{1}{2}}V^7}{C_2 L^5}$.

S = area of wetted surface in square feet.
L = combined length of entrance and run in feet.
L_a = Length of run in feet.
A = area of immersed midship section in square feet.
V = speed of ship in knots of 6,080 feet.
h = depth of immersion of axis of propeller in feet.

$\varepsilon = 0.56$	$\log \varepsilon = \bar{1}.748188$
$e = 2.71828$	$\log e = 0.434290$
$2\pi = 6.2832$	$\log 2\pi = 0.798180$
$C_1 = 32{,}566$ for clean paint.	$\log C_1 = 4.512761$

C_2 and C_3 are constants, depending upon the form of the ship, C_3 having a provisional value 1.66

ANALYSIS OF POWER.

If from the indicated horse-power, the power required to overcome the constant resistance of the unloaded engine is deducted, the result will be the net horse-power applied to propulsion; and the ratio of P_t to this net horse-power will represent the combined efficiency of the mechanism of the engine so far as the variable friction is involved, and of the propeller, including the effect of the frictional wake and of the augmentation of resistance by the propeller. These latter two effects are thus assumed proportional to the power applied. The ratio $\frac{P_t}{Net\ H.P.}$ may then be expected to remain constant so long as the propeller efficiency remains constant.

APPLICATIONS.

The results of the progressive trial of the U. S. S. *Bancroft* given by Passed Assistant Engineer Robert S. Griffin, U. S. Navy,* suffice for the determination of the value of the constant C_2 for that vessel. The *Bancroft* is a twin-screw practice ship of the following dimensions: length of immersed body $= L = 189.5$ ft.; displacement on trial $= 832$ tons; area of immersed midship section on trial $= A = 277$ sq. ft.; wetted surface on trial $= S = 6{,}150$ sq. ft.

Having calculated the power absorbed by surface friction on the basis of Froude's coefficient and exponent, and also the value of the factor involving the following current, the constant C_2 is given such a value as will maintain the ratio $\frac{P_t}{Net\ H.P.}$ approximately constant. The value of C_2 thus found is for the *Bancroft* 329, and the ratio $\frac{P_t}{Net\ H.P.} = 0.545$. The resistance of the unloaded engine is assumed at 2 lbs. per square inch of the area of the low-pressure pistons. Using these values, the power at the several speeds has been calculated and is given in Table I, where the observed powers are also given to show the degree of approximation.

TABLE I.—U. S. S. *BANCROFT*.

V	4.85	6.80	8.60	10.40	11.95	13 30	14.52
P_t	16.55	43.50	87.42	161 63	266.70	409.33	601.60
Net H.P.	30.37	79.82	160.40	296.56	489.35	751 06	1,103.86
I.H.P. by formula.	53.09	109.82	197.90	341.56	541.85	811.06	1,171.36
I.H.P. by trial	65.00	110.00	200.00	340 00	540.00	815.00	1,170.00

In order to illustrate the scope of the formula the value of the ratio $\frac{P_t}{Net\ H.P.}$ is given in Table II together with explanatory data for each of seven vessels, five of them being twin-screw ships of the U. S. Navy, and the two others the S. S. *Umbria* of the Cunard line and the torpedo boat *Sunderland*, built by Messrs. William Doxford and Sons, Sunderland, England. With the

*Journal of Society of Naval Engineers, Vol. V, p. 384.

TABLE II.

SHOWING THE VALUE OF THE RATIO $\frac{P_t}{Net\ H.P.}$ FOR SHIPS OF VARIOUS TYPES AND SIZES.

Data are for trial conditions.	Combined length of entrance and run. L	Area of immersed midship section. A	Area of wetted surface. S	Displacement in tons of 2,240 lbs.	Depth of immersion of axis of propeller. h	$\frac{L}{\sqrt{A}}$	Propelling engines only. $I.H.P.$	$Net\ H.P.$	Speed in knots of 6,080 feet. V	$\frac{P_t}{Net\ H.P.}$	Reference.
Bancroft	189.5	277	6,150	832	7.54	11.39	1,170	1,102	14.52	0.5457	Jour. Am. Soc. Naval Eng. Vol. V, p. 365.
Machias	193	324.6	6,816	1,067.5	8.00	10.71	1,794	1,693	15.46	0.5500	*Ibid* Vol. V, p. 846.
Philadelphia..	332	815	20,100	4.325	11.38	11.63	8,533	7,972	19.68	0.5860	*Ibid* Vol. II, p. 524.
New York.....	384	1,350	30,814	8,480	10.50	10.45	16,947	16,019	21.co	0.5705	*Ibid* Vol. V, p. 613.
Umbria..	352*	1,000	35,061	9,860	11.00	11.13	13,920	13,136	20.00	0.5600	*Ibid* Vol. III, p.293.
Vesuvius.....	251.7	192	7,545	771	6.25	18.17	3,896	3,699	21.18	0.5814	*Ibid* Vol. III, p.582.
Sunderland...	137	39.93	†1,664	86	3.70	21.68	232	200	14.04	0.5855	Engineering (London), Vol. XLIX, p. 32.

* The length of immersed body of the *Umbria* is 500 feet. The length $L = 352$ feet is obtained by ascribing to the combined entrance and run a prismatic coefficient of fineness of 0.56.

† This value of the wetted surface of the *Sunderland* is the product of $LA^{\frac{1}{2}}$ by a constant derived from the U S. Torpedo Boat *Cushing:* see Journal American Society of Naval Engineers, Vol. IV, p. 1.

exception of the *Umbria* these vessels may be regarded as approximations to the form of least resistance in smooth water. The *Umbria* has a quasi-parallel middle body, the length of which has been approximated by ascribing to the combined entrance and run a prismatic coefficient of fineness of 0.56. Table III contains the results of the progressive trial of the *Umbria* together with the calculated power. The same comparison is shown graphically in Fig. 1.

TABLE III.—S. S. *UMBRIA*.

V	8	10	12	14	16	18	20
I.H.P. by formula	1,008	1,698	2 692	4,123	6,210	9,302	13,920
I.H.P. by trial	1,050	1,600	2,400	3,800	6,050	9,420	13,920

In making these calculations, the constants of the *Bancroft* have been employed, their values being $C_1 = 32{,}566$, $C_2 = 329$, $C_3 = 1.66$. It should be stated that the application of this value of C_2 to all of these vessels is not strictly warranted, for they are undoubtedly dissimilar in cross section and also in mode of distribution of displacement. Nevertheless it is assumed that they possess a sufficient degree of similarity to serve the purpose of illustration.

The resistance of the unloaded engine is based on two pounds per square inch of the area of the low-pressure pistons in all cases. The length L is the combined length of entrance and run, being the length of the immersed body in all cases except the *Umbria*. The ratio $\frac{L}{A^{\frac{1}{3}}}$ is given as an index to the character of the ship from a propulsive point of view. If all these vessels possessed the required degree of similarity, and had propellers and engines of equal mechanical efficiency, the ratio $\frac{P_t}{Net\ H.P.}$ should be the same for all, provided the speeds were within the limit indicated.

LIMIT OF SPEED.

Since the formula gives only the power absorbed in surface friction and in waves generated without interference, it is obvious

that it may be used as a measure of the total power only when these elements constitute the entire resistance. The earliest departure from the condition of applicability may be expected to accompany an interference by the recurring transverse bow wave in the generation of the stern wave. Since the relative magnitude of the bow and stern waves is dependent upon the form of the ship, it is clearly impossible to formulate generally the limit at which such an interference will occur. The limit must therefore be found from the performance of the model ship whence the constants in any case are derived.

The U. S. S. *Bancroft*, within a speed of 14.52 knots, gives no evidence of the limit, as may be seen by reference to Table I. At this speed $V^2 = 1.112\ L$.

The S. S. *Umbria* gives no indication of the limit within the speed of 20 knots, as may be seen by Table III or Fig. 1. At 20 knots $V^2 = 1.136\ L = 0.8 \times$ length of ship.

On the other hand, the progressive trial of the torpedo boat *Sunderland* affords a convenient illustration of the inapplicability of the formula. Figure 2 shows the power curves, both actual and calculated. At the speed of 14 knots the value of the ratio $\frac{P_t}{Net\ H.P.}$ is such as to warrant the conclusion that at this speed the formula gives the power with sufficient accuracy. A glance at the curves, however, shows that this cannot be said of any higher speed until 20 knots is approached. Fourteen knots may then be regarded as the limit of speed at which the formula may be applied. At this speed $V^2 = 1.43\ L$. Beyond this speed the familiar hump appears. It may be of interest to note that the curves intersect again in the region of 20 knots, at which $V^2 = 2.9\ L$.

It is not intended to question the existence of minor humps at speeds below the limits here noticed, but simply to observe that in the published curves of the *Bancroft* and *Umbria* they are not appreciable. It is possible that such humps may be partially neutralized by the action of a simultaneous increase in the wake current upon the propeller. In any event their existence must

affect the accuracy of the calculated power, although commonly to such a slight extent as to be inappreciable.

Since the magnitude of the waves of the ship varies with the ratio $\frac{A^{\frac{1}{2}}}{L}$ it follows that both the size of the hump and the speed at which it appears will be influenced by this ratio.

CONCLUSION.

The method of this analysis may be summarized thus: The resistance of ships is so largely due to surface friction and wave making that all other forms may be neglected except at very high speeds. Within the limit of speed at which interference occurs in the generation of waves, the formula (eq. 8) enables the power absorbed by these two principal elements to be computed, provided the constants C_2 and C_3 have been experimentally determined by means of a model ship having the same mode of distribution of displacement as the projected ship. The ratio of the power thus computed to the net power of the engine is constant so long as the propeller efficiency is constant. Finally, the indicated horse power may be obtained by adding to the net power the power absorbed in the constant resistance of the engine.

APPENDIX.

PROBLEM.—Given the speed V and the displacement D; required the length L of combined entrance and run, which will make $P_f + P_w$ a minimum.

Let D = displacement in tons of 2,240 lbs.,

t = a constant, such that tL = length of immersed body,

f = prismatic coefficient of fineness $= \frac{35D}{tLA}$,

g = constant value of ratio $\frac{S}{tLA^{\frac{1}{2}}}$.

Then

$$A = \frac{35D}{ftL} \text{ and } S = gtLA^{\frac{1}{2}},$$

whence $$P_f + P_w = \frac{gV^{2.83}}{C_1}\left[\frac{35DtL}{f}\right]^{\frac{1}{2}} + \frac{V^7}{C_2}\left[\frac{35D}{ftL^3}\right]^{\frac{3}{2}},$$

which becomes a minimum when

$$L^5 = \frac{315 C_1 D V^{4.17}}{C_2 g f t^2}. \qquad (1)$$

Example.—A ship similar to the *Bancroft* in cross section and distribution of displacement is to have $D = 832$ and $V = 10$; required the length L at which $P_f + P_w$ will be a minimum.

For the *Bancroft*, $t =$ unity; $g = 1.95$; $f = 0.5548$; and $C_2 = 329$.

For a clean condition of bottom $C_1 = 32{,}566$ and eq. (1) gives

$$L = 204.$$

For a moderately foul bottom, say $C_1 = 25{,}000$, the length becomes

$$L = 193.$$

PROBLEM.—Given the speed V and the area of the midship section A; required the length L of the combined entrance and run which will make $P_f + P_w$ a minimum.

As before $S = gtLA^{\frac{1}{2}}$,

and $$P_f + P_w = \frac{gtLA^{\frac{1}{2}}V^{2.83}}{C_1} + \frac{A^{\frac{3}{2}}V^7}{C_2 L^3},$$

which becomes a minimum when

$$L^4 = \frac{3 C_1 A V^{4.17}}{C_2 g t}. \qquad (2)$$

Example.—A ship similar to the *Bancroft* in cross section and distribution of displacement is to have $A = 277$ and $V = 14$; required the length L which will make $P_f + P_w$ a minimum.

$t =$ unity; $g = 1.95$; $C_2 = 329$; $C_1 = 25{,}000$.

By eq. (2). $L = 210$.

That is to say, a ship having the same midship section as the *Bancroft* and a length of 210 feet would require less power at 14 knots than the existing ship, and would have a displacement about one-ninth greater.

PROBLEM:—Given all dimensions and constants appropriate to the ship, and the ratio $\frac{P_f}{P_w} = r$; required V.

S.S. U

Power Curves

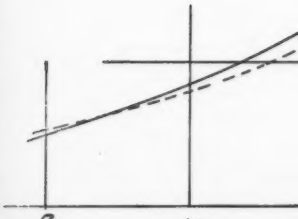
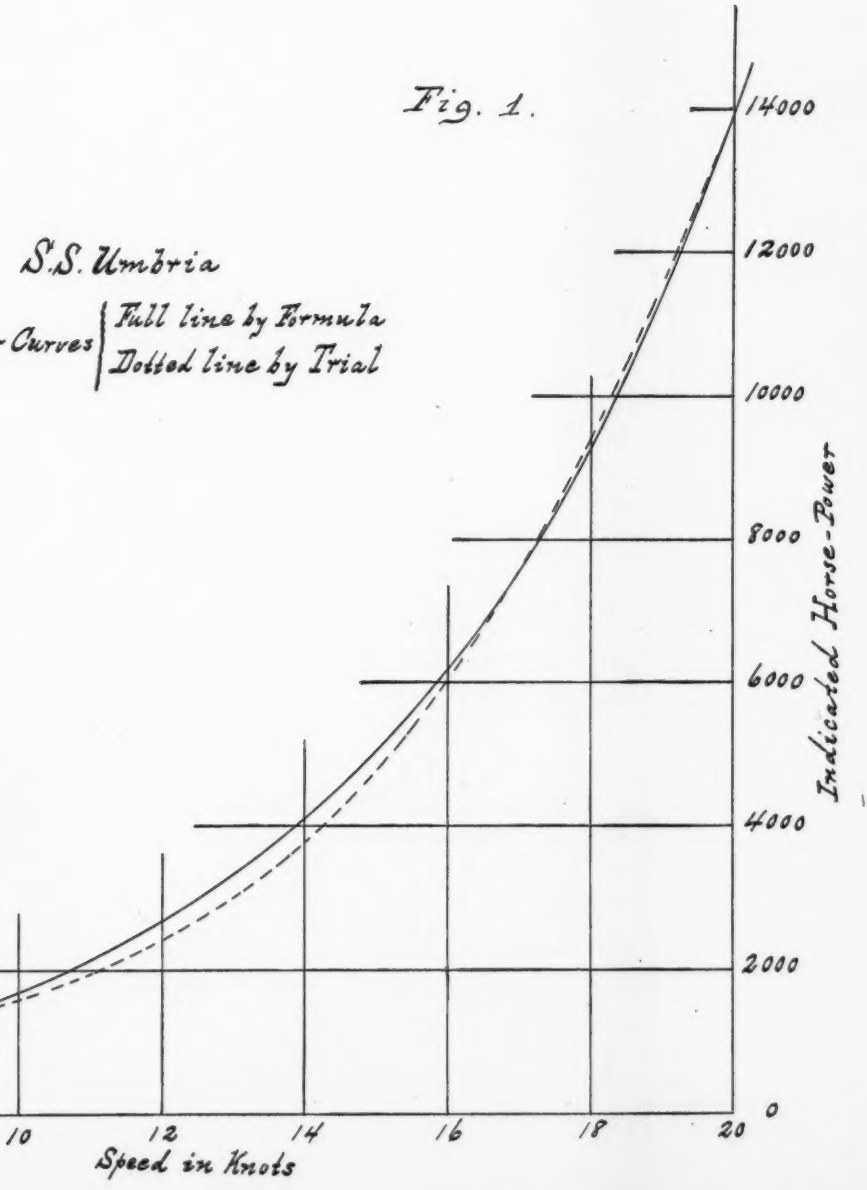


Fig. 1.


S.S. Umbria
Curves
Full line by Formula
Dotted line by Trial
14000
12000
10000
8000
6000
4000
2000
0
Indicated Horse-Power
10
12
14
16
18
20
Speed in Knots

Torpedo Boat

Power Curves

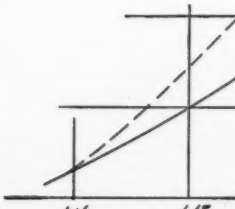
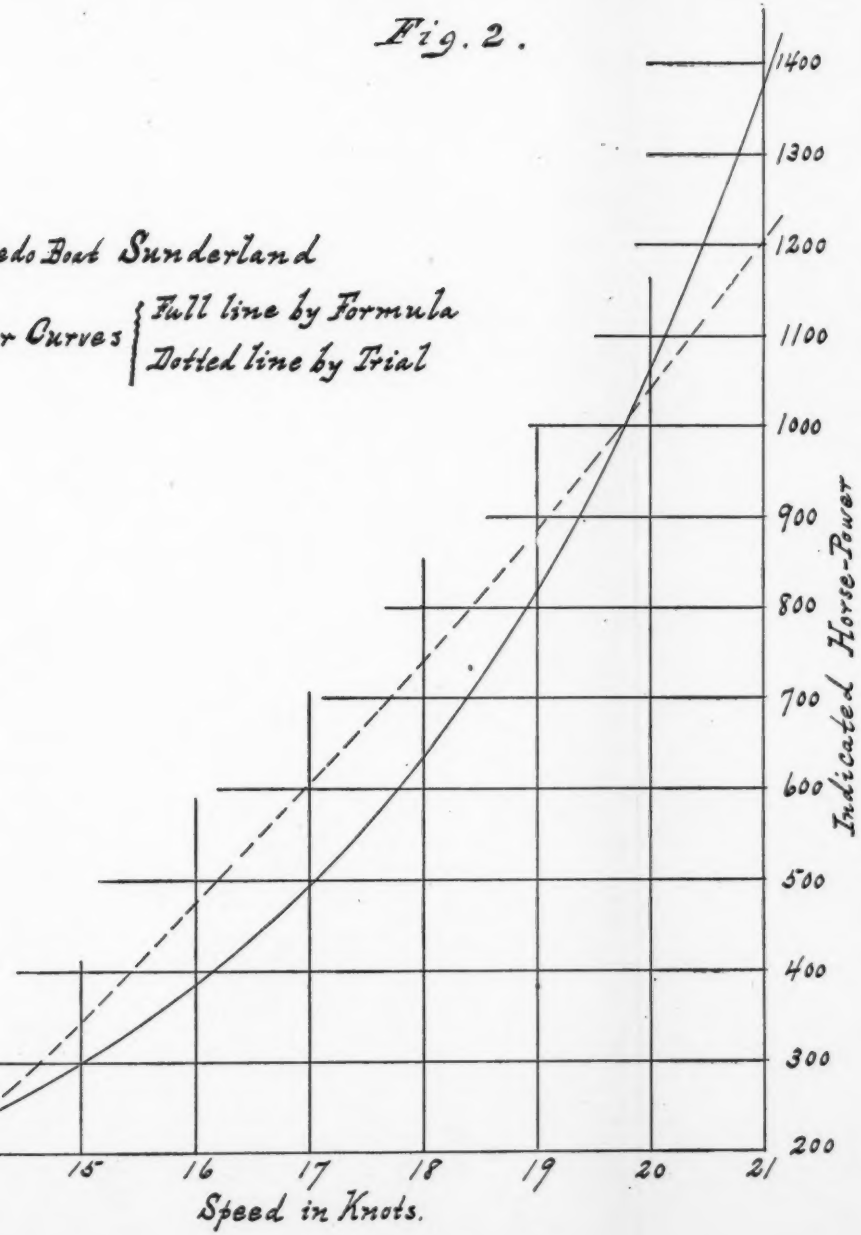


Fig. 2.


edo Boat Sunderland
r Curves
Full line by Formula
Dotted line by Trial
1400
1300
1200
1100
1000
900
800
700
600
500
400
300
200
Indicated Horse-Power
15
16
17
18
19
20
21
Speed in Knots.

$$r = \frac{P_f}{P_w} = \frac{SC_2L^3}{C_1A^{\frac{1}{2}}V^{4.17}}.$$

Whence $$V^{4.17} = \frac{SC_2L^3}{rC_1A^{\frac{1}{2}}}. \qquad (3)$$

Corollary:—Since $S = gtLA^{\frac{1}{2}}$, eq. (3) may be written

$$V^{4.17} = \frac{C_2gtL^4}{rC_1A}.$$

Whence $$L^4 = \frac{rC_1AV^{4.17}}{C_2gt}. \qquad (4)$$

When $r = 3$, eq. (4) becomes identical with eq. (2).

Hence for a given area of midship section, when the length is such as to make $P_f + P_w$ a minimum, the ratio $r = 3$.

ON THE INERTIA STRESSES OF A SLIDE VALVE.

By John H. Macalpine, Esq., Associate.

The principal object of the present paper is the consideration of a special problem in kinematics which presented itself naturally in the course of practical work.

In examining the valve motion of a large and high-speed engine it became evident that it did not exactly correspond to the type known as simple harmonic—that is, very approximately, such a motion as the valve would have were it actuated by a single eccentric, and an eccentric rod very long in comparison with the valve travel. As piston valves were adopted in the case of all cylinders, the stresses on the valve gear were due almost entirely to the inertia of the valves. As these stresses were of considerable magnitude it was desirable to determine them with accuracy.

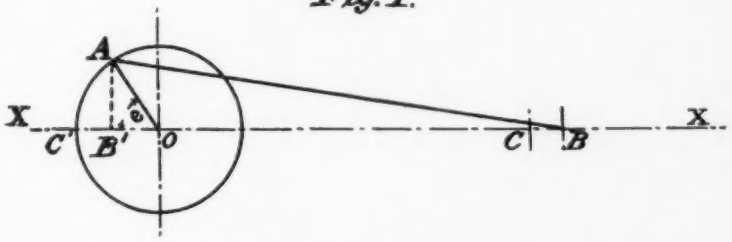
The possible effects of elasticity in the various parts of the gear were also examined, as several parts of it were subjected to cross-bending stresses, but at present we will only deal with those accelerations of the valve motion, and consequent stresses, which would exist if each part of the gear were rigid—that is, with the acceleration and stresses deduced from a valve ellipse which had been got at geometrically from a diagram of the gear.

The slight departures from simple harmonic motion mentioned above are not such as have any importance when considering steam distribution. That they may be important when considering the strength and stiffness of the gear can be easily shown.

Let us begin, *ab initio*, with the case of a valve, or any mass actuated by a single eccentric, where the eccentric rod is attached direct to the valve spindle, and is long in comparison with the length of travel of the valve—the usual arrangement in an ordinary non-reversible engine.

The acceleration and inertia stresses of the valve can be very easily calculated in this case, since the movement of the valve is very nearly simple harmonic, the crank shaft being supposed to rotate at a uniform angular velocity.

Fig. 1.



In Fig. I, O is the center of the crank shaft, XX the center line of the valve spindle, and AB the eccentric rod. B, the joint of the valve spindle and eccentric rod, has of course the same motion as every point of the valve. C is the position which B occupies when nearest O; that is, when OA occupies the position OC' in line with XX. If AB' is drawn perpendicular to XX, the length $C'B'$ will only differ from CB by a very small and varying quantity introduced by AB being inclined at the small angle ABO to XX. The motion of B' is exactly simple harmonic (since it fulfills the definition of this kind of motion), and consequently that of B very nearly so.

Since B' is always directly under A, the motion of B' is the resolved part of the motion of A parallel to XX. Similarly, the acceleration of B' is the resolved part of the acceleration of A parallel to XX.

If we call V the linear velocity of A in feet per second, and R the radius OA in feet, it is well known that the acceleration of A is equal to $\frac{V^2}{R}$ in feet per second, and that it is directed towards O. Thus the acceleration of B'

$$= \frac{V^2}{R} \times \frac{OB'}{OA} = \frac{V^2}{R^2} \times r,$$

where $r = OB'$ in feet.

Denoting by ω the angular velocity of the radius OA in radians* per second, we have $\frac{V}{R} = \omega$, and thus the acceleration of $B' = \omega^2 r$.

The stress required to give a mass m this acceleration is $\frac{m\omega^2 r}{g}$ in gravitation units, and is always directed towards the center of motion.

Thus the stress has a maximum value $\frac{m\omega^2 R}{g}$, which it reaches when OA coincides with XX, and it is seen to be proportional to the travel of the valve and the square of the revolutions per minute of the engine conjointly.

From Fig. I we see then that the motion of the point B is nearly the same as that of B', but not quite. There is a slight disturbance caused by the two elements of the gear OA and OB. In fact the maximum stress of the valve, instead of being $\frac{m\omega^2 R}{g}$ as given above for a mass m having the same motion as the point B', is

$$\frac{m\omega^2 R}{g}\left(1 + \frac{OA}{AB}\right).$$

If we pass now to any of the ordinary reversible gears, we have a considerably greater number of parts introduced, the disposition and dimensions of each of which must influence the valve motion. The effects of the various parts in causing it to depart from simple harmonic motion may be to some extent compensatory. Indeed in almost all gears in which an attachment is made to the connecting rod, such as Joy's, Kirk's or Morton's gears, several parts are introduced specially to produce such a compensation, and thus prevent the valve motion from being very unequal with respect to the two ends of the cylinder, as it would be were a simple direct connection made to the connecting rod.

Still, it will be found on examination that the valve motion shows a very sensible departure from simple harmonic motion;

* The radian is the unit angle in circular measure. It is the angle subtended by a circular arc equal in length to the radius, $= 57°\text{-}17'\text{-}45''$.

but this is usually more pronounced in the case of non-eccentric gears than in that of eccentric gears, no doubt, principally on account of the large angles through which some of the parts of the former gears have to move. The problem of assigning to each part its exact effect in the valve motion we may very well leave till mathematicians have supplied us with a complete solution of the three-link linkage.

Again, the importance of these disturbances often increases in a very rapid ratio as the valve travel is increased for the same length of engine. This we might expect, since the dimensions of the gear must remain about the same, while the increase of valve travel is got by increase of motion (angular, linear, or both combined) of some at least of the parts.

Thus, while the disturbances we are considering may safely be overlooked in, for instance, the ordinary merchant marine work, their neglect may lead to an underestimate of the stresses, sometimes to the extent of even 50 per cent. in the designing of warship engines, for here the engines must be short and of high power; consequently the gear is more or less cramped, the valves heavy and the valve stroke long. But in all high-speed design it is trouble well spent to examine this point.

To show the importance which these small irregularities may have, we will revert to our simple case in Fig. 1.

If θ is the angle which OA makes with XX, the displacement of B' from O is given by the equation

$$r = R \cos \theta.$$

And if t is the time which has elapsed since OA coincided with XX,

$$\theta = \omega t \text{ and } r = R \cos \omega t.$$

The time T of a complete vibration of B' (that is, the time occupied by B' from its leaving C' till it again reaches it) is evidently given by

$$\omega T = 2\pi, \text{ or } T = \frac{2\pi}{\omega},$$

as r would always have the same value at intervals of time which were equal to $\frac{2\pi}{\omega}$, or multiples of this value.

This motion is represented in Fig. 2 by the full curve AB, which, however, shows only the motion for one-quarter of the complete period. The times t are measured along the base line OB, while the ordinates of the curve give the values of

$$r = R \cos \omega t. \quad OA = R.$$

Similarly, $$r_1 = R_1 \cos n\omega t$$

represents a simple harmonic motion of which R_1 is the maximum displacement of the moving point from its mean position; but in this case the time of a complete vibration is seen, by comparison with the equation of the first motion, to be $\frac{2\pi}{n\omega}$, that is, only $\frac{1}{n}$th of what it was in the first case. This motion we may represent by such a curve as CDE, *etc.*

In the motion given by

$$r = R \cos \omega t \text{ (curve } AB),$$

the acceleration at time t was

$$\omega^2 r = \omega^2 R \cos \omega t,$$

and the stress $= \dfrac{m\omega^2 R \cos \omega t}{g}$.

Similarly when $r_1 = R_1 \cos n\omega t$ (curve CDE, *etc.*), we have the acceleration $= n^2\omega^2 R_1 \cos n\omega t$, and the stress $= \dfrac{mn^2\omega^2 R_1 \cos n\omega t}{g}$.

This may be seen by comparison of corresponding quantities in the case of the curve AB.

Now, suppose a point to move so that its displacement at any time is the sum of the displacements in the two simple harmonic motions given above. That is

$$r_2 = r + r_1 = R \cos \omega t + R_1 \cos n\omega t.$$

This motion is represented in Fig. 2 by the curve FGH, *etc.*

The acceleration is, of course,

$$R\omega^2 \cos \omega t + R_1 n^2 \omega^2 \cos n\omega t.$$

If R_1 is small compared with R, the new motion differs but slightly from our case in Fig. 1, the maximum value of r_2 being $R + R_1$. But the maximum value of the acceleration is

$$\omega^2 (R + n^2 R_1);$$

or, in other words, while the maximum disturbance of the displacement has only been in the ratio

$$1 + \frac{R_1}{R},$$

the acceleration and also the stresses have been altered in the ratio

$$1 + n^2 \frac{R_1}{R}.$$

This is clearly brought out by comparing Figs. 2 and 3, where corresponding curves are designated by the same letters. Those in Fig. 2 give the displacements as explained, and in Fig. 3 the ordinates give the accelerations and stresses, each measured to the proper scale. We notice at once that, while the two curves AB are of precisely the same form, the corresponding ordinates of the curves FGH differ very markedly.

Thus if n be comparatively large, and consequently the second motion be of short period compared with the first, the change of acceleration and of stress may be of importance, although the actual displacement at any time is but slightly changed from that due to the first simple harmonic motion.

In Figs. 2 and 3, $n = 6$, and $\frac{R}{R_1} = 80$.

The importance of the illustration will be appreciated when we recollect that any periodic motion can be divided into a set of simple harmonic motions of various periods and amplitudes, the longest period being the same as that of the complete motion, as was first shown by Fourier, early in the century, in his great work on the conduction of heat.

We pass now to the case of an actual valve motion.

To determine the inertia stresses we must first know exactly the motion of the valve. This is usually represented diagrammatically in the following manner:

Suppose any straight line AB to represent the length of the stroke of the engine. Let the distance (in the direction of motion of the valve) of any arbitrarily-chosen point of the valve, from any arbitrarily-chosen point on the valve face, be noted for a sufficient number of positions of the piston, and draw a

curve as follows: When the piston has reached a point Q (Fig. 4), suppose the distance apart of the above points to be y. Draw a perpendicular to AB at Q, and lay off $PQ = y$. Joining all the points thus found will give a closed curve, which usually does not differ greatly from an ellipse, and is called the valve ellipse.

The most obvious procedure by which to determine the inertia stresses of the valve, is to use a method which has been known since the time of Newton. That is, to determine the velocity curve, and from that to derive the curve of accelerations. For our present purpose we need only give a very slight sketch of this method.

Divide the crank circle into an equal number of parts. These equal parts will represent equal lines, as the crank is supposed to revolve uniformly. If we find the position of the piston for each of those points, we can determine the displacement of the valve from the valve ellipse. Lay off a straight line AB, Fig. 5, equal in length to the circumference of the crank circle, and divide it into the same number of equal parts. Then construct a curve by laying off the displacement of the valve at each of those points at right angles to the base line AB, in a similar manner to our procedure in the case of the valve ellipse.

Knowing the revolutions at which our engine is supposed to run, we can at once find what distance CD will be traversed in one second, or fraction of a second, by the crank pin. We now draw CE tangent to the curve at C. This can, of course, be done most exactly by determining the direction of the normal at C by means of a pair of trammels, one end being so placed that the other will move along the curve in the proximity of C. Now CE is the line which would have represented the valve motion had the velocity of the valve been uniform. It would have reached the point E, vertically above D, in one second. Thus, at uniform velocity the valve would have increased its displacement by a distance DE. DE, therefore, measures the velocity of the valve at the point C.

From the displacement curve we can thus find the velocity of the valve at any number of points, and lay down a complete curve of velocities on the base AB.

Treating tne curve of velocities in precisely the same way as we have treated the curve of displacements to deduce the velocity curve, we can determine the curve of acceleration, and thus determine the inertia stresses of the valve.

This construction arises at once from the fundamental ideas of velocity, as the time rate of change of position of a body, and of acceleration, as the time rate of change of velocity. We have, in fact, at the very root of the calculus and kinematics the same expressions $\frac{dy}{dt}$ and $\frac{dv}{dt}$, giving on the one hand the trigonometrical tangents of the angles, which the geometrical tangents to the displacement and velocity curves at any point make with the base line; and, on the other hand, denoting the velocity and acceleration at the point.

Fig. 6 shows the result of an actual case, where the curves were drawn to a large scale, and each curve determined with all possible accuracy from nearly a hundred points.

A glance at the displacement curve of this figure shows at once that we have here a very considerable departure from simple harmonic motion, and this is confirmed by the velocity and acceleration curves. If the displacements had given a pure, simple harmonic curve, all three—the displacement, velocity, and acceleration curves—would have been of precisely the same nature, each being a curve of sines. This is far from being the case, though the departure is greater here, probably, than very frequently happens. The whole motion could, by a well known method (which it would be entirely apart from the purpose of this paper to explain here), be analyzed into a series of simple harmonic motions, the period of the longest one being the same as the period of revolution of the engine, and the periods of the others exact submultiples of this, each having its own proper amplitude. A pair of such motions have been combined in Figs. 2 and 3, and it has been shown how much more marked is the effect of the short period motions on the acceleration than on the displacement curve. That such short-period disturbances are present in the actual valve motion shown in Fig. 6 we have the clearest evidence of in the acceleration curve of this figure.

But it will readily be seen that a very slight error in determining the inclination of a tangent to the displacement curve will give a large error in the curve of accelerations; for such an error exactly corresponds to an actual irregularity of very short period in the displacement curve. Thus, although every precaution was taken in determining the form of the acceleration curve in Fig. 6, it is difficult to say whether some of the minor irregularities have not been exaggerated or partly obliterated by small errors in the process. Indeed it was to a great extent the doubt as to the nature of the result in Fig. 6 which led to the search for the new method shortly to be explained.

Besides the disadvantages in the method just given, the process is a long and tedious one.

We will now explain a process by which the inertia stresses can easily and quickly be determined direct from the valve ellipse. This method only involves two measurements of the valve ellipse for each point at which the stress is to be determined: One, the measurement of the inclination of the curve at the point to the base line from which the displacements of the valve are marked off. As will be shown further on, a slight error in this measurement has no great effect at those points at which the valve inertia stresses are a maximum. The other measurement is the length of the radius of curvature of the valve ellipse, which, if the curve is carefully laid off to a proper scale, can be measured within two or three per cent. of error. This measurement has also the advantage that it is more easily made with accuracy where the acceleration of the valve is high, as at those times the radius of curvature of the valve ellipse is smaller than it would be for a lower acceleration.

We believe that this is the first time that this method has been made public. But a paper on it formed one of three papers privately circulated by the author in 1889, for a special purpose.

We may think of the valve ellipse, Fig. 6, as being formed in the following manner:

Suppose the point Q to move, carrying the perpendicular PQ with it, along the base line AB in such a manner as to keep pace with the piston; and the point P to move along this per-

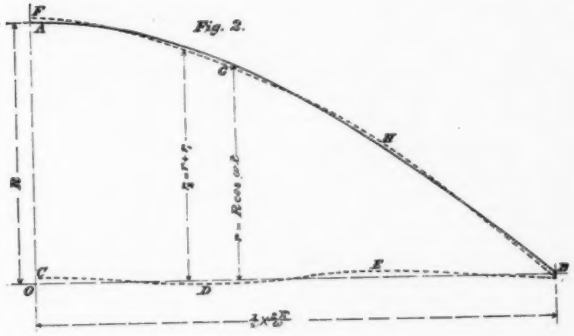


Fig. 2.

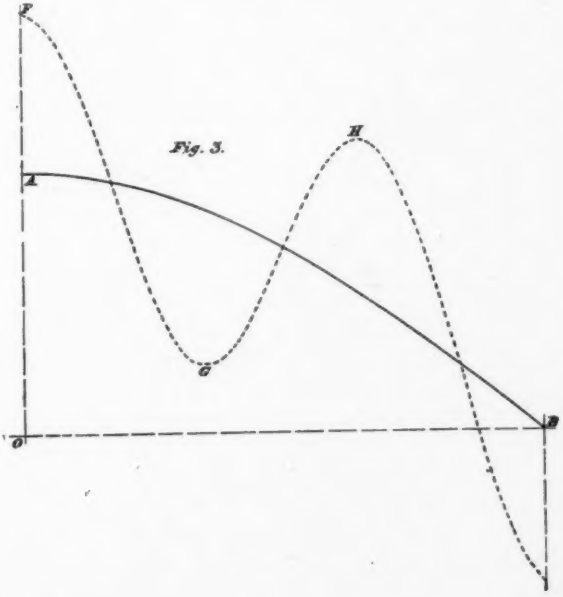


Fig. 3.

Fig. 4.

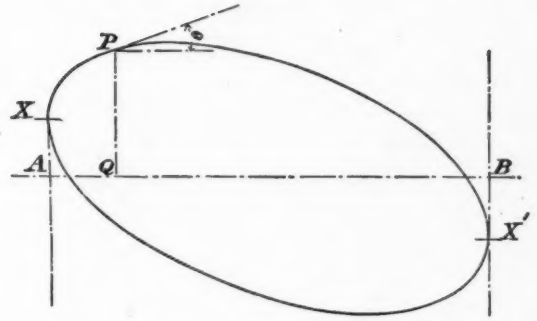


Fig. 5.

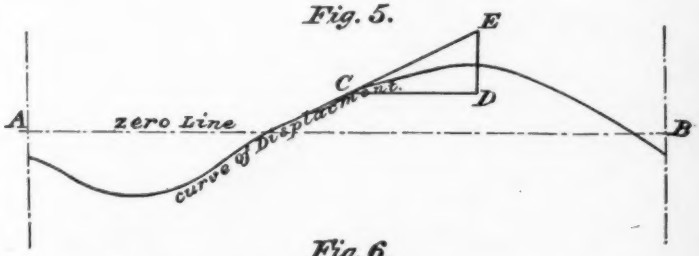


Fig. 6.

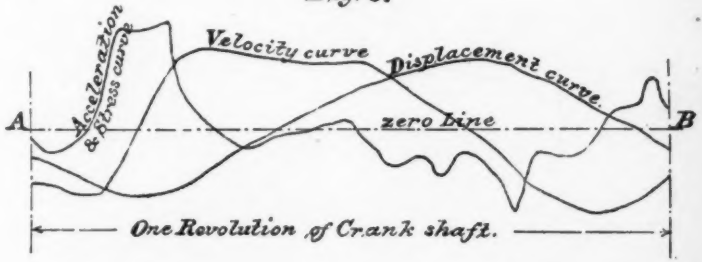


Fig. 7.

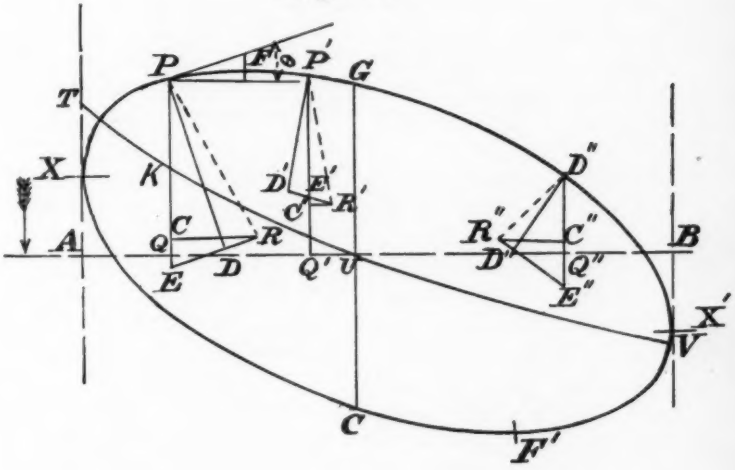
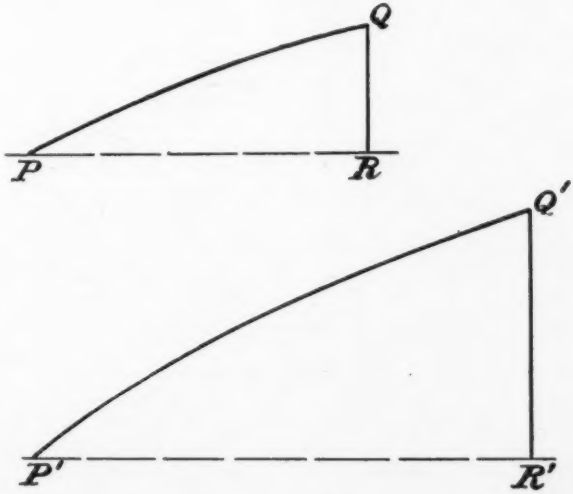


Fig. 8.



pendicular so as to keep pace with the valve. P will trace the valve ellipse. The acceleration and velocity of P parallel to AB are the same as those of Q, that is, of the piston. But the acceleration and velocity of P perpendicular to AB are the same as those of the valve. The proportion of these quantities at any time will determine the radius of curvature of the valve ellipse at the point, and its inclination to AB. The measurement of those two quantities from the curve, together with the known velocity and acceleration of the piston at all times, are then sufficient to determine the velocity and acceleration of the valve.

The relation between the various quantities may be established very simply. Fig. 7, which is a repetition of Fig. 4 with some additions, is lettered in the same manner. Let PR represent in direction and magnitude the total acceleration of the point P. This, as has just been explained, is made up of the acceleration $PC = \beta$, of the valve; and the acceleration $CR = \alpha$, of the piston when at Q. We may also resolve the acceleration PR into the acceleration PD normal, and the acceleration DR tangential to the valve ellipse at P. Produce RD to meet PQ at E.

If v represents the velocity of the piston at Q, and V the velocity of the point P along the curve

$$v = V\cos\theta, \text{ or } V = \frac{v}{\cos\theta},$$

where θ is the inclination of the tangent at P to AB, as shown in the figure. Then V can be determined when θ has been measured from the curves.

The normal acceleration of P is $\frac{V^2}{\rho}$, ρ being the radius of curvature of the curve at the point P. Thus, since the angle $DPC = \theta$ and PDE is a right angle,

$$PE = \frac{V^2}{\rho\cos\theta} = \frac{v^2}{\rho\cos^3\theta},$$

also

$$CE = CR\tan\theta = \alpha\tan\theta.$$

Now, as the figure is drawn, the acceleration PC of the valve is the difference of those two quantities PE and CE, and if we call an acceleration of P positive which is in the direction of the

arrow, and the acceleration of the piston positive when it is from A towards B, we get the following equation for the acceleration of the valve,

$$\beta = \frac{v^2}{\rho \cos^3 \theta} - \alpha \tan \theta. \qquad (1)$$

We need here say nothing of the quantities v^2 and α, as methods of determining them are quite generally known.

The problem we have been dealing with, and of which equation (1) is the solution, has been explained with reference to its connection with the valve ellipse. But it could obviously have been treated quite apart from any question of engineering, and is purely a problem in kinematics which may be stated thus:

A point Q, Fig. 7, moves along AB in any manner, the law of which is known; and P, always perpendicularly above Q, describes a known curve. What relation has that part of the acceleration of P which is perpendicular to AB, to the known velocity and acceleration of Q, and the curvature and inclination of the curve described?

We will now give the answer to various questions which have arisen when the method was being used.

First, we will follow the point P, round the valve ellipse in order to see clearly the change of sign which the two terms on the right-hand side of equation (1) undergo. In the first term v^2 is, of course, always positive. Also, ρ will not change sign unless the center of curvature changes from one side of the curve to the other. This it would only do in any actual case where the curve had a point of contrary flexure. In Fig. 7 it never changes sign, as it is obviously always finite, and we have written the equation on the assumption that its value is always positive. In other words, let a line be drawn through any point of the curve to the center of curvature for that point, and produced indefinitely through the center of curvature, ρ will be positive if this line cuts the curve again, and it will be negative if the line lies altogether outside the curve. $\text{Cos}^3\theta$ is positive for the portion of the curve XPX', and is negative for the other half.

Altogether, we have then the first term, $\frac{v^2}{\rho \cos^3 \theta}$ positive for the portion XPX' of the curve, and negative for the other portion.

We will deal now with the second term, $-\alpha \tan \theta$. The ordinates of the curve TUV give the values of α, QK representing its value at the point Q. α is positive from A to U, at which point TUV cuts the line AB; it is then negative from U to B and back again to V. It then changes to positive, and remains so till A is again reached. Tan θ is positive from X to F, the highest point of the curve, negative from F to X', positive from X' to F', the lowest point of the curve, and again negative from F' to X.

Thus, on the whole, we have $-\alpha \tan \theta$

negative from X to F,
positive from F to G,
negative from G to X',
positive from X' to F',
negative from F' to G',
positive from G' to X.

The values of the terms, and whether they should be added or subtracted, can also be seen from the construction for the acceleration β at the points P, P', P'', &c., which require no further explanation, as the lettering corresponds all through.

As the greatest values of the accelerations β will occur very near F and F', the turning points of the valve motion, the values of θ for these maximum values of β will be small, and the second term of equation (1) of little importance. Also, $\cos \theta$ differs very slightly from unity. Thus, as we stated above, a slight error in the measurement of θ has little effect in the value of the maximum acceleration deduced from the curves.

In the foregoing we have always supposed AB to be the actual length of the stroke of the engine, but this is not usually so. It is clear that v^2 and α must be determined, at each point, for the same revolutions as the actual engine is supposed to run at, but for a length of stroke equal to AB, not to the stroke of the engine.

There are several points as to the scale to which the curves are drawn which should be mentioned. If the *same* valve motion is represented by two valve ellipses which have different

scales for AB, the stroke of the engine, and y, the displacement of the valve, the accelerations of the valve deduced should be the same in both cases. It is not quite apparent at first sight that this will be so. Suppose, for instance, that the valve motion was set off full size, and that AB was also the actual length of stroke of the engine. Suppose, again, that the same valve motion was set off c times full size, on a base AB which was drawn k times full size, c and k being greater or less than unity. If no correction is made for scale, then in this last case the acceleration deduced should be c times that deduced from the first case, but it should be independent of k.

In Fig. 8, let PQ and $P'Q'$ represent corresponding parts of the curves drawn to the different scales. Also, let all the quantities be designated by the same letters as we have used previously, as θ,a, &c.; but in the second case with accents added, as θ', a', &c. Thus

$$P'R' = k \times PR \text{ and } Q'R' = c \times QR.$$

If the arcs PQ and $P'Q'$ be supposed exceedingly small,

$$\tan \theta = \frac{QR}{PR} \text{ and } \tan \theta' = \frac{Q'R'}{P'R'} = \frac{c}{k} \times \frac{QR}{PR} = \frac{c}{k} \tan \theta.$$

a, the acceleration of the piston, will be proportional to AB; thus

$$a' = ka$$

Therefore, on the whole, we have

$$a' \tan \theta' = ka \times \frac{c}{k} \tan \theta = ca \tan \theta,$$

which shows the correctness of the second term of the formula in this respect.

The changes that take place in the first term are not so obvious. If φ and φ' are the angles of inclination to the base lines at Q and Q', the arcs PQ and $P'Q'$ still being supposed small, and ρ and ρ' the radii of curvature of those arcs, we get

$$\tan \theta' = \frac{c}{k} \tan \theta, \text{ and } \tan \varphi' = \frac{c}{k} \tan \varphi.$$

Therefore $\tan \varphi' - \tan \theta' = \frac{c}{k} (\tan \varphi - \tan \theta)$.

But $$\tan\varphi - \tan\theta = \frac{\sin\varphi\cos\theta - \cos\varphi\sin\theta}{\cos\varphi\cos\theta}.$$

$$= \frac{\sin(\varphi-\theta)}{\cos\varphi\cos\theta} = \frac{\varphi-\theta}{\cos^2\theta}$$

very approximately, since φ and θ differ only slightly.

Similarly, $$\tan\varphi' - \tan\theta' = \frac{\varphi'-\theta'}{\cos^2\theta'}.$$

Thus, $$\frac{\varphi'-\theta'}{\cos^2\theta'} = \frac{c}{k}\times\frac{\varphi-\theta}{\cos^2\theta}.$$

Also, $$PQ = \frac{PR}{\cos\theta}, \text{ and } P'Q' = \frac{P'R'}{\cos\theta'};$$

$$\rho = \frac{PQ}{\theta-\varphi} = \frac{PR}{(\theta-\varphi)\cos\theta}, \quad \rho\cos^3\theta = PR\times\frac{\cos^2\theta}{\theta-\varphi},$$

and $$\rho'\cos^3\theta' = P'R'\times\frac{\cos^2\theta'}{\theta'-\varphi'} = kPR\times\frac{k}{c}\left(\frac{\cos^2\theta}{\theta-\varphi}\right)$$

$$= \frac{k^2}{c}PR\frac{\cos^2\theta}{\theta-\varphi} = \frac{k^2}{c}\rho\cos^3\theta.$$

Next $$v'^2 = k^2v^2$$

Thus for the first term we have

$$\frac{v'^2}{\rho'\cos^3\theta'} = k^2v^2 \div \frac{k^2}{c}\rho\cos^3\theta = c\times\frac{v^2}{\rho\cos^3\theta}.$$

This corresponds, as it should, with what we have found for the second term.

The fact that the accelerations of the valve given by the formula are proportional to the scale of displacements, or in other words to the stroke of the valve, and independent of the scale of AB, might of course at once be inferred. But it will no doubt give a clearer view of the structure of the formula to follow the changes of scale in detail.

On the subject of the scales we would only make one further remark, as to their choice. θ can always be determined with sufficient accuracy whatever the scales of the valve ellipse, but

it is different with ρ. To measure ρ accurately it must not be very large. We have found from experience that the measurements may be well made if AB is about two feet long while the valve travel is drawn to a scale which will give it a length of from nine inches to one foot. It is clearly necessary that the curve be drawn with great accuracy and sharpness.

The formula (1) fails at the two points X and X'. It is not usually important to find the values of β or the stresses at those points, but we will show how it may be done. Putting $v^2 = V^2 \cos^2\theta$, V being as before the velocity of the point P along the curve, equation (1) becomes

$$\beta = \frac{V^2 \cos^2\theta}{\rho \cos^3\theta} - a \tan\theta = \frac{\frac{V^2}{\rho} - a \sin\theta}{\cos\theta}. \qquad (2)$$

$\frac{V^2}{\rho}$ is the normal acceleration of P which, at X and X', is equal to a. Sin $\theta = 1$ at those points, so that both numerator and denominator of the above fraction vanish and it assumes the form $\frac{0}{0}$. Treating this fraction in the usual way, by differentiating numerator and denominator separately, we get (differentiating (2) with respect to y).

$$\beta = \frac{\frac{2V}{\rho}\frac{dV}{dy} - \frac{V^2}{\rho^2}\frac{d\rho}{dy} - a\cos\theta\frac{d\theta}{dy} - \sin\theta\frac{da}{dy}}{-\sin\theta\frac{d\theta}{dy}}, \qquad (3)$$

Now at X, $V = \frac{dy}{dt}$, so that

$V\frac{dV}{dy} = \frac{dV}{dt} = -\beta$, as the positive direction of β has been chosen downwards.

Cos $\theta = 0$, and $\frac{da}{dy} = o$, as a has reached a maximum value at this point.

Lastly, $$\frac{d\theta}{dy} = \frac{1}{\rho}, \text{ and } \frac{V^2}{\rho} = a.$$

Putting these values in (3) we get

$$\beta = \frac{-\frac{2\beta}{\rho} - \frac{a}{\rho}\frac{d\rho}{dy}}{-\frac{1}{\rho}} = 2\beta + a\frac{d\rho}{dy},$$

or $$\beta = -a\frac{d\rho}{dy}.$$

At X', $$\beta = +a\frac{d\rho}{dy}.$$

To evaluate $\frac{d\rho}{dy}$ at X and X' the valve ellipse near those points must be determined from a greater number of points than usual. Then, by carefully measuring ρ for several points between X or X' and the first tenth of the stroke on either side, and also noting the corresponding value of y for those points, we may construct a curve with y for the abscissæ and ρ for the ordinates. The fact that we should obtain thus a fairly smooth curve will serve as a check on the accuracy of the measurements of ρ. All we have to do then is to draw a tangent at that part of this curve corresponding to the points X and X' on the valve ellipse. The trigonometrical tangent of its inclination to the line of abscissæ is the value $\frac{d\rho}{dy}$. It is evident that $\frac{d\rho}{dy}$ will vanish for points near X and X'; for, as we pass along the curve in either direction from those two points we soon come to parts at which ρ is manifestly increasing.

Of course, at the points X and X', we could also proceed by drawing out the displacement, velocity, and acceleration curves on a time base and thus determine the stresses.

Finally, it may not be uninteresting to add the analytical deduction of equation (1), as it is naturally this method rather than the geometrical one given above which one uses in investigating such a question.

Representing AQ, Fig. 7, by x and using all through the same notation as above, the quantity we wish to express is $\beta = -\frac{d^2y}{dt^2}$.

Now
$$\frac{dy}{dt} = \frac{dy}{dx} \cdot \frac{dx}{dt}.$$

And
$$\frac{d^2y}{dt^2} = \frac{d}{dt}\left(\frac{dy}{dx}\right)\frac{dx}{dt} + \frac{dy}{dx} \cdot \frac{d^2x}{dt^2} = \frac{d^2y}{dx^2}\left(\frac{dx}{dt}\right)^2 + \frac{dy}{dx} \cdot \frac{d^2x}{dt^2}. \quad (4)$$

$$\frac{dx}{dt} = v, \quad \frac{d^2x}{dt^2} = a, \text{ and } \frac{dy}{dx} = \tan\theta.$$

$$\frac{1}{\rho} = \frac{-\frac{d^2y}{dx^2}}{\left\{1 + \left(\frac{dy}{dx}\right)^2\right\}^{\frac{3}{2}}} = \frac{-\frac{d^2y}{dx^2}}{\left\{1 + \tan^2\theta\right\}^{\frac{3}{2}}} = -\cos^3\theta\,\frac{d^2y}{dx^2},$$

or
$$\frac{d^2y}{dx^2} = -\frac{1}{\rho\cos^3\theta}.$$

Substituting these quantities in equation (4) we get

$$\beta = -\frac{d^2y}{dt^2} = \frac{v^2}{\rho\cos^3\theta} - a\tan\theta.$$

This is equation (1).

The changes of scale are also very easily followed in this way. Pursuing the same notation as above, we have

$$y' = cy \qquad x' = kx,$$

$$\frac{dy'}{dx'} = \frac{dy'}{dy} \cdot \frac{dx}{dx'} \cdot \frac{dy}{dx} = \frac{c}{k}\frac{dy}{dx},$$

$$\frac{d^2y'}{dx'^2} = \frac{d}{dx}\left(\frac{dy'}{dx'}\right)\frac{dx}{dx'} = \frac{c}{k}\frac{d^2y}{dx^2} \cdot \frac{1}{k} = \frac{c}{k^2}\frac{d^2y}{dx^2}.$$

That is
$$\frac{\mathrm{I}}{\rho' \cos^3 \theta'} = \frac{c}{k^2} \frac{\mathrm{I}}{\rho \cos^3 \theta},$$

$$v' = \frac{dx'}{dt} = k \frac{dx}{dt} = kv,$$

$$a' = \frac{dv'}{dt} = k \frac{dv}{dt} = ka,$$

$$\tan \theta' = \frac{dy'}{dx'} = \frac{c}{k} \frac{dy}{dx} = \frac{c}{k} \tan \theta.$$

Substituting these we finally get

$$\beta' = c\beta \text{ as before.}$$

THE ACCIDENT TO THE *CITY OF PARIS.*

By Otto H. Mueller.

[Translated by permission from the Zeitschrift des Vereines Deutscher Ingenieure, by William Wachsmann, Associate.]

The official investigation into the cause of the accident to the *City of Paris*, on March 25, 1890, continued from the 16th to the 24th of June, 1890, and, although the testimony of not less than twenty-eight witnesses and experts was taken, concluded by acquitting all parties concerned, of any blame, and at the same time declared that the construction, material, and design of the vessel and machinery were faultless, and that the conduct of the entire ship's company, including the engineer's force, was deserving only the highest praise.

It further declared that all opinions previously held concerning the cause of the accident would not hold, though the real cause was not found; from which it would follow that human foresight could not prevent a similar accident to another vessel at any time.

Those who have followed the proceedings of the Board of Trade in similar cases (*e. g.*, the accident on board the *Barracouta*, in February, 1890,) expected little in this instance; and that the result should have been such a negative one has met with general disapproval, and English engineering journals ("Engineer" and even the local papers of Liverpool) say that the case is not settled by this verdict, but that it should be brought before another tribunal, however disagreeable it might be for the parties concerned. This would, no doubt, be very difficult; for up to this time it has been no easy task for parties not concerned to gain access to the vessel and the ruins of machinery; besides, the latter had long since been forwarded to Glasgow.

However, before reporting on what I personally saw and heard in Liverpool from the 2d to the 9th of May, 1890, I think it would be well to describe the vessel and machinery, based upon reliable data, and partly upon personal observation.

I.

The *City of Paris*, belonging to the Inman and International Steamship Line of Liverpool, is (with the sister ship, *City of New York*) one of the largest and fastest of ocean steamers. Her length on the water line is 527 feet, on deck 563 feet; the greatest breadth 63 feet 3 inches; depth from top of upper deck to keel, about 42 feet 9 inches; draught, measured amidships, 24 feet and over, according to load, the displacement varying from 10,000 to 13,000 tons. The full coal supply for one voyage from Liverpool to New York is 2,500 tons, sufficient in itself to cause an increase of 4 feet in draught. The double bottom has a depth of from 3 to 4 feet, and has a capacity of 1,600 tons of water ballast.

She has four continuous decks running from stem to stern (see Figs. 1 and 2); they are the lower deck, the main deck, the upper deck, and the promenade deck; and below the lower deck, in the forward and after parts of the ship, is the orlop deck and a platform deck. Amidships, instead of these two decks, there are two continuous stringers so as to retain longitudinal connection. The general height from deck to deck is about 8 feet, but from the upper to the promenade deck it is about 9 feet, and from the platform to the orlop deck about 7 feet 6 inches. In the forward part of the vessel, on the two lower decks, are the provision rooms, ice room, cable tier, and chain lockers. In the after part, the space adjoining the run or stern of the vessel serves as cargo space; it has a capacity of 2,000 tons, but is never used during the height of the traveling season, as it takes too much time to load and unload in port.

The lower deck, amidships, for the full length of the engine and boiler space, contains nothing but coal bunkers; forward and aft of this, and above the water line, are cabins for first and

second-class passengers, the after of which are in compartments 11 and 12, and were for the most part under water after the catastrophe. Abaft and adjacent to these are a number of cabins for second-class passengers, while the third-class passengers are provided for in the extreme forward and after parts of the ship. The main deck is similarly divided. On the upper deck, forward, are the lavatories, closets, and hospital for third-class passengers. Then follows the dining saloon for first-class passengers, which on the deck above extends the full length of the promenade deck. The central portion is 22 feet high, 25 broad, and 53 long, the whole being covered by an arched stained-glass skylight one and one-half inches thick. The entire saloon is 70 feet long and about 50 feet wide, and will seat about 250 people. Adjoining this saloon, aft, is the dining room for children, officers' and engineers' mess, lavatories for ladies and gentlemen, and last of all a large smoking room. Immediately adjoining the latter, is the dining room for second-class passengers; then follow the cabins for officers, and in the stern the hospital for first and second-class passengers. Above the center of the upper deck, which is 49 feet wide, is the promenade deck, 29 feet wide, at the forward part of which is the arched skylight of the dining saloon, and abaft this the ladies' saloon and the reading room. She can accommodate 1,500 passengers of all classes, and the accommodations for the first-class ones are equal to those to be found in any modern first-class hotel.

The promenade deck on either side of the cabins is 17 feet wide; upon it are located twenty-six folding life boats.

The central portion of the ship below the upper deck, for a length of 160 feet, is entirely taken up by the engines, boilers and coal bunkers.

Hydraulic power has been introduced everywhere, in order to avoid the unpleasant noises caused by the chains of steam winches used in loading and unloading freight. The officers and crew number 375, of whom 176 belong to the engineers' force alone, viz: 1 chief engineer, 18 assistant engineers, 6 of whom hold certificates as first-class engineers, 3 for dynamos and 3 for ice

machines, 3 for feed pumps, 33 oilers, 54 firemen, 57 coal passers and 1 yeoman.

The vessel is a three-master, and although the sail area is 11,000 square feet, it is not capable of imparting any forward motion to the ship. It consists chiefly of fore-and aft sail, and is intended merely to steady her in a sea-way. She was built in 1888 and 1889 by James and George Thomson, of Clydebank (Glasgow), in accordance with the rules of Lloyds and the Board of Trade; at the same time, the British Admiralty requirements were satisfied, so that in case of war she might be used as a cruiser. All the engines, boilers and appurtenances were made by James and George Thomson, the dynamo, ice and pumping machinery only being of special make.

The weight of the hull itself is about 7,000 tons, and the material is Scotch steel. Fourteen water-tight bulkheads divide the vessel into fifteen compartments. They extend up to the main deck, and are $\frac{3}{8}$ inch thick below and $\frac{}{16}$ inch above the water line. There is, besides, a water-tight bulkhead between the engines; near the center of it there is a water-tight door, stiffened by $3\frac{1}{2}$ by 7-inch Z-bars.

The *City of Paris* started on her first voyage to New York on the 3d of April, 1889, and up to the time of the accident had made nine such voyages. During August of that year she made the fastest trip that was ever made between Sandy Hook and Queenstown, viz: 5 days, 19 hours and 18 minutes, which is equivalent to a speed of 20.6 knots, or 23.75 miles per hour. Up to that time the *Etruria* and the *Umbria* were considered the fastest vessels afloat, their record being 6 days, 2 hours and 7 minutes.

II.

She has twin screws, each of which is driven by a vertical, inverted, triple-expansion engine of 8,000 to 10,000 horse power, the plan and arrangement of which is clearly shown on Figs. 1 to 5. The steam pressure varies from 140 to 150 pounds per square inch. The high-pressure cylinders are 45 inches in diameter, the intermediate 71, and the low-pressure 113; the stroke

is 5 feet. The cylinder liners are made of cast iron and are jacketed. The cylinder covers and heads are not jacketed; there is no means of doing it, as there are no openings in the ribs, as could be seen on the broken cylinder cover. The surface thus exposed in the L. P. cylinder amounted to 48 per cent. of the surface of the cylinder. The jackets are heated by live steam from the boilers, the pressure of which is regulated by means of a reducing valve. The drainage of the jackets is attended to by the engineers opening from time to time a cock, *h* (Fig. 3), on a water trap provided with a glass gauge. The valve chests and passages were drained in a similar manner, so that in all there were six such traps, all of which discharged into the air-pump pressure space. I was informed that the jackets consumed from 4 to 5 per cent. of all the steam produced. (The low-pressure jacket was originally intended to take steam and condensed water from the other jackets, and this water was then to pass into the cylinder to be again converted into steam. It appears, however, that this scheme did not prove a success. The entire exhaust steam from the auxiliary machinery, amounting to about 500 I.H.P., is passed into the low-pressure steam chest.)

All the cylinders are fitted with piston valves, the high-pressure having one, the intermediate two and the low-pressure four, all of about 24 inches diameter. They receive their motion from eccentrics secured to the coupling flanges of the crank shafts, and are reversed by the usual Stephenson link motion. The eccentrics are 54 inches in diameter, and have a throw of about 8 inches. In the high-pressure cylinder the steam is taken between the valves, and in the others at the ends. Balance pistons are provided for all valve stems to carry the weight of the valve, rods and eccentric straps.

The pistons are of cast steel. The rod is secured to the piston by a flange and eight bolts 3¼ inches in diameter, instead of the usual taper and nut, as shown in Fig. 6, which is a copy of J. and G. Thomson's working drawing. This peculiar design appears to have been adopted solely at the request of the engineer-in-chief of the Inman Line, Mr. J. Doran, a fact I ascertained during a conversation with that gentleman. He claimed that a

great many pistons have been split by setting up on the nut where the tapered piston rod was employed, and for that reason he was in favor of a flanged joint. This could only happen to such pistons where the taper had very little draw. Such a case has never come under my notice when the taper had a good draw, nor do I know of it having occurred with anybody else. In fact the tapered piston rod end is used by all leading marine-engine builders, and has also been used on the latest engines of the United States Navy. A more detailed description of this piston will be given hereafter.

The cylinders are not connected by faced flanges, but as usual, by two rods *bb*, about 3 inches in diameter (see Figs. 3 and 4), the flanges *a'a''* being smooth, but not faced. They are further connected by rods *ff*. They rest on cast-steel inverted Y-frames, the latter resting on a bed plate of the same material. The fore-and-aft distance of the piston valves of the L.P. cylinder from center to center is about 11 feet 6 inches, so that their distance from the bearing surfaces *vv* is about 4 feet 6 inches. The low-pressure cylinders only have tail rods. On the high and intermediate they were cut off and the stuffing boxes closed by blank flanges, so as to save weight, lubricant and packing materials, and to give about 240 additional I.H.P. The engines were fitted with a Dunlop governor, which acts only when the ship is pitching, but will not prevent racing in a smooth sea. It is remarkable that many English engineers and designers claim that it is impossible to construct a governor that will prevent an engine racing under all conditions, and that it is neither practicable nor necessary with triple-expansion engines with cranks at 120 degrees apart, on account of the regularity of turning. The average number of revolutions were 80 to 84, equivalent to a piston speed of 800 to 840 feet, but in the original design only 75 revolutions were proposed. Cross-heads, guides and connecting rods were built in the usual manner. (See Figs. 3 and 4.) The crank shaft was in three similar sections, each section in five pieces, making fifteen pieces in all. With the exception of the cylinders and pistons, every part of the engine was interchangeable; one piston valve, one piston rod, one crosshead, one eccentric

and rod, one crank shaft, one bed plate, etc., will answer as spare parts for all six cylinders of both engines. The bed plate is not cast hollow, but is open on top, is ribbed and has large cored openings; it appears weak in spite of considerable depth, though the same could not be said of the engine foundation. I had only a hurried glance at the foundation plan in the engineering office of the Inman company, but I remember that the keelsons were at least 4 feet 2 inches high, the angle irons about 8 by 9 inches, and the distance between angle irons under bed plate not more than 12 inches. The drawing of the foundation in Figs. 3 and 4 is to be accepted only as a general scheme; the center of the shaft might be about 9 inches too high above the bottom of the ship.

The surface condensers are cylindrical and are made of brass. Their location and that of the air pump, is shown in Figs. 3 and 5. At *n*, near one end of the condenser is the centrifugal pump and engine; at *m* two vertical Worthington feed pumps; at *l* two vertical fire and bilge pumps; at *k*, a large vertical accumulator steam pump of about 100 I.H.P., without a fly wheel; it is regulated automatically according to the needs of the hydraulic machinery, one half of which is supplied by each accumulator.

There are the following auxiliary engines in the vessel: In the engine rooms, two accumulator pumps, four feed pumps, four fire and bilge pumps, and two circulating pumps; in addition to this there is a steam cylinder for each reversing engine and another for opening the main stop valves; abaft the engine rooms, four compound dynamo engines, with cylinders $7\frac{1}{2}$ and 13 inches in diameter and 12 inches stroke, and two ice machines with cylinders 6 inches diameter and 12 inches stroke; over the boilers, twelve fan engines for forced draft; there are also two steam anchor-hoisting engines; so that in all there are 32 auxiliary engines in the ship. The fans for ventilating the passenger quarters are driven by electric motors, each fan being capable of discharging 4,000 cubic feet of air per minute. The dynamos supply about 1,000 incandescent lights. Hydraulic power is used for turning over the main engines (the driving wheel *w* can be seen on the after flange of the thrust shaft in Fig.

5), for the steering engines, for the eight ash hoists, for an engine-room elevator, four kitchen elevators, one feed hoist for the stables, and 12 cargo whips, each of 30 tons capacity.

Access to the engines is had by three platforms. On the lowest (No. 1, Fig. 3) all the cranks and bearings can be overlooked. The gauges are above the hand wheel d', and near it is a lever e' for opening the steam cylinder for the main stop valve. The second platform gives access to the crossheads, guides, links, eccentric bearings, air-pump connections, stuffing boxes, etc. The third platform is for the oilers alone. On the upper side of the intermediate cylinder there are two brass oil boxes about 20 inches long, 10 inches wide and 16 inches deep, from which the oil is led through copper tubes to the different connections on the second platform.

Contrary to the usual practice the 5½-inch bolts for holding down the caps of the main bearings are not in the center, but about three inches out of center and towards the crank pins (see Fig. 5). Each line of shafting consists of ten pieces connected by flanges, with fourteen bearings and one thrust bearing. Only the 42-foot length of outboard shafting and the three sections of crank shaft are hollow; all are of Vickers' steel.

The outboard bearing is shown in Figs. 7 and 8. It is cast in one piece with the upper strut, and the lower strut is connected to the bearing by means of a flange; both parts are of cast steel, and weigh 26 tons. The lignum vitæ was not secured in the usual way with a rib separating each strip, but the strips were laid so as to touch each other, and were only prevented from turning by a feather at the top. They were secured by wrought-iron bands driven in tight over the ends, and were kept in place by a ring secured to the bars.

The boss of this bearing was connected to the side of the ship by a conical sheet-iron casing about 60 feet long, and the casing itself with the skin of the ship by the plate struts *tt*, Fig. 1. Although this method of stiffening the outboard bearing appears very solid, it is objectionable from the fact that for about 60 feet of its length the shaft is inaccessible. In case it is necessary to inspect the screw shaft journal it becomes necessary to remove

the screw and tear away a considerable portion of the casing. The large plate struts moderate the pitching of the vessel.

The screw propellers are three-bladed, and are made of manganese bronze; originally they were made of steel.

In regard to the performance of the engines, I found that at 16 knots in very bad weather (head wind) it took 13,700 I.H.P., and at 20 knots in good weather 18,800; the power ordinarily developed is 16,000 I.H.P.

III.

Forward of the engines, in three water-tight compartments, each with a smoke pipe, are the nine double-ended boilers, with a total heating surface of 50,265 square feet. Each boiler is 15 feet 6 inches in diameter, and 19 feet long; has six furnaces, each 47 inches diameter, and 1,056 tubes, $2\frac{3}{8}$ inches outside diameter, and 7 feet 6 inches long. The grate bars are from 1 to $1\frac{3}{16}$ inch thick. The boilers have a common combustion chamber. The tops of the three oval smoke pipes are about 100 feet above the center furnaces. The boilers are always worked with forced draft, on the closed stokehold system; otherwise they would not be capable of furnishing enough steam, for the ratio of the inside diameter of the tubes to their length is 1 to 41, compared with 1 to 25, to 1 to 30 for boilers with natural draft. The air pressure in the fire rooms is kept at from $\frac{7}{16}$ to 1 inch of water; generally $\frac{1}{2}$ inch. The blowers take air from the deck. The temperature in the fire rooms varies from 95 to 120 degrees Fahr., and the evaporation with Welsh coal is about 9.5 pounds. The average coal consumption is 330 tons a day, which seems rather high for the average. For the return voyage from New York, Pennsylvania (bituminous) coal is used. In order to preserve the boilers, the feed water is heated to 200 Fahr. by a special heater, supplied with steam from the boilers. The vacuum is 26 inches, with a temperature of injection of 50 Fahr. From these figures, the calculated steam consumption per I.H.P. per hour is 15.4 pounds, exactly the same as that of the *Meteor*, also by J. and G. Thomson, and the evaporation per square foot of

heating surface 5.5 pounds; not much more than is obtained from boilers with natural draft.

The total weight of boilers, including water, is about 2,000 tons; that of the entire machinery, including propeller shafting, bearings, propellers and struts for supporting the outboard bearings, over 1,000 tons; to this must be added 2,500 tons of coal, making in all 5,500 to 6,000 tons for boilers, engines and coal.

IV.

Annually, during January, the vessels of the Inman Line are docked for the inspection, cleaning and repairing of the hull and machinery. The *City of Paris* was docked on the 6th of January, 1890, in drydock No. 3, at Birkenhead, and remained there until the 18th of January. On the 5th of February she was to have left for New York. Several defects were discovered in the engines, and were made good. Cracks were found in the columns, one of which had been repaired in New York sometime before. An inspection of the starboard outboard bearing showed a wear of $\frac{5}{16}$ inch, and the upper play in the lignum vitæ was $\frac{1}{2}$ inch. The bearing on the port side showed no evidence of wear. This was also the case with the *City of New York*, which had been running eight months longer than the *City of Paris*. The repairs made consisted in fitting new lignum vitæ in the lower half of the bearing without taking out the shaft, which would have taken about fourteen days more, although the foreman of Laird Brothers had orders to make repairs regardless of expense.

The next voyage to New York was made without mishap, as was also the following one. On the last (ninth) voyage she left New York March 19th, 1890, with 687 passengers and 2,000 bales of cotton. On the 23d the starboard engine had to be stopped for twenty minutes to pack the high-pressure stuffing box. From that time to the afternoon of the 25th everything went well until about 5·20 P. M., when the captain and some others noticed large volumes of steam issuing from the starboard-engine skylight, and immediately afterwards felt heavy repeated shocks on the vessel.

The oiler. Cunningham, who went on watch at 4 o'clock on platform III, claims to have observed no racing of the engine, which may be very true, if we consider that a racing engine, with total absence of pressure on the bearings, runs almost noiselessly, with the exception of the pumps; but these being so far below platform III, and there being little steam to condense, probably worked very quietly also, and during the general noise the varying gait of the starboard engine from that of the port, which was running at a quiet gait, could not be easily detected. Shortly before this, this oiler had wiped off the low-pressure tail rod, had stepped off the cylinder cover, and was just in the act of cleaning one of the valve-chest bonnets, when he suddenly saw pieces of the cylinder head fly up and steam escape which burnt him. He took refuge in one of the store rooms on platform III, in order to protect himself from the hail of iron that was flying around, and afterwards crept to one of the ladders, which he had hardly reached when he heard a tremendous crash, and then made his escape.

Below, on platform I, was third engineer Gill, who held a first-class certificate, and whose watch commenced at 5·20. He states that at this time the steam gauges registered 140 pounds, and that the engine was making from 81 to 82 revolutions, and that he found all bearings running cool, and then went over to the port engine, closing the door in the bulkhead behind him. Immediately afterwards he heard the starboard engine begin to race, and hurried back and put the link in mid position, as near as he could, with the engine room full of steam. While thus engaged a large piece of iron fell close to him, and he made his escape to the port engine room.

These statements seem very doubtful. If Gill really wanted to stop the engine he could have done so easier with the handle e′, which actuates the steam cylinder for closing the main stop valve, and where he would have been much better protected from the "iron hail," for this handle is about six feet further away from the low-pressure cylinder than the reversing wheel. Furthermore, how was it possible for the engine to raise the weight of 55 tons of shafting if the links were in mid position, or nearly

so? During his further testimony, he said that the engine room was filled with steam later on, and that he was engaged in pumping out the port engine room until midnight. The other witnesses declared that the height of the water forced them to retreat about 11 o'clock. At 8 o'clock the water was 22 feet deep.

Fifth engineer Holcroft relieved McLaren at 5·15, and felt the bearings of the port engine. He was afterwards engaged in examining condensed water in the starboard engine room, when he suddenly perceived that the starboard engine was racing; he concluded that the screw was broken, and then made his escape.

Engineer Gibson had the watch from 12 o'clock noon until 4 P. M., but returned again to the engine room at 5·20 to relieve McLaren, so that he could go get his tea. (Holcroft had already relieved McLaren.) He felt the bearings of the starboard engine and everything seemed to be working well, the steam pressure being 145 pounds. While he was feeling the bearings of the port engine, the starboard engine began to race, and a heavy piece came flying through the bulkhead behind him (see holes *LL* in bulkead, Figs. 3 to 5.) What it was he could not say; he neither looked for it, nor did he find anything. He ran back to the port engine room, and went up to the store room for a light, and was afterwards engaged in closing the doors in the bulkheads and in pumping out. As he looked into the starboard engine room, he saw the low-pressure cylinder falling down, after which all lights were extinguished.

The other engineers, examined by the court, Carnegie and Walls, were not in the engine room during the accident, but were in their cabins. If there was anybody else below besides the four men above mentioned, their testimony certainly would have been taken by the court. Therefore, it must be concluded that there were only four engineers present in both engine rooms, while a full watch should have consisted of at least twelve engineers and oilers. And of these four, Gibson and Holcroft, did not belong there; their places should have been filled by McLaren. Where the others were the Liverpool court did not in-

quire; but according to the testimony of engineer Walls, who was busy filling out the starboard engine room journal, in his cabin immediately above the starboard engine, he heard noise below, but on account of the steam could not reach the starboard engine, but reached it by means of the firemen's entrance on the main deck, and observed about eight different engineers there idle, and ordered them to work.

McLaren states that he was in the engine room about 5 o'clock, and observed nothing unusual, while he was relieved by Holcroft at 5·15, and by Gibson at 5·20.

It was the intention of the builders to keep the door in the bulkhead always closed. Mr. Biles, the former naval architect of J. and G. Thomson, and the designer of the *City of Paris*, stated before the court that this door was put there solely, or mainly, to control the water in the two engine rooms. At the Inman office I was informed that the personnel of the two engine rooms was an entirely separate one; that the one does not bother about the other, and for that reason Mr. Biles doubted that Gill could have walked from the port to the starboard engine and put the valve gear in mid-position.

After having thoroughly read the proceedings of the court, I am forced to admit that I can accept the statements of only one of the four eye witnesses of the accident, viz: that of the oiler, Cunningham (except that of Holcroft, who doubtless told the truth when he said that he made his escape without even having attempted to stop the engine.)

Let us now follow up the further occurrences based on the testimony of John Walls, the chief engineer, whom I personally met, and whose statements I accept as being perfectly true.

When Walls finally gained access to the stop valve of the starboard engine, with the assistance of another engineer he closed it, then hurried back to the main deck, and from there to the port engine, and closed the valve there also. Here he met engineer Carnegie, who, after the catastrophe, hurried below, and from him he first learned what had happened, and which he had failed to see on account of the steam in the engine room, viz: that the low-pressure cylinder, with the columns, air pump, and

everything connected with it had disappeared; that the connection of the condenser with the sea was destroyed, and that in that way the water was rapidly filling the engine room. Thereupon, the door in the longitudinal bulkheads was closed, the port engine bilge pumps started, and Walls and an assistant tried to close the sea connections in the starboard engine room (19 in all), but were only successful with a few. The others were so badly damaged that to try to close them would have been useless. In the meantime, the vessel fell into a heavy sea, and rolled so heavily—20 degrees to starboard and 23 degrees to port—that Walls and his companions, now up to their necks in water, had to give up all further attempts. Considering further that the entire engine room was in darkness—it was also getting dark outside—and that the inrushing water made every free movement impossible, that Walls had to feel and climb his way around the ruins, the water gaining every second, it must be admitted that this man did his duty. A few moments later the water rushed from the starboard to the port engine room, there being a number of holes in the longitudinal bulkhead about 12 inches diameter, and 8 feet above platform I. All attempts to close these holes with wedges and mattresses proved futile, the shock of the waves being too heavy. In the meantime, the carpenter reported that water had made its appearance in compartments Nos. 11 and 12 (Fig. 1), and was rising rapidly. By effective pumping it was possible to partially save these compartments, so that upon arrival at Queenstown the water in No. 12 was pumped down to twelve feet, and in No. 11 to six or six and a half feet. These and other compartments in the ship had no hand pumps at all, and were only connected with the engine bilge pumps by suction pipes. After these were destroyed, the water flowed from the engine room through the suction pipes to compartments 11 and 12. Here is a case where the bilge pumps served to flood the compartments instead of draining them. The cabins for first-class passengers on the lower deck, which have already been referred to, were in these compartments.

The bulkhead No. 9, between engine and boiler room, had already sprung several inches, and threatened to give way on

account of the force of the waves. Walls supported it with wooden props placed against the boilers, and this prevented any further trouble from that side. The captain tried to prevent any more water from coming in through the sea valves by drawing sails around the ship, but it was of no avail, as the heavy seas tore the sail away again and again. Night came on. In the engine compartments and in the dynamo room (between Nos. 9 and 10) the water was about 25 feet deep, and just about as deep in compartments 11 and 12. Over 3,000 tons of water was already in the ship!

Notwithstanding this there was still 11 feet freeboard, and therefore no danger of going down. Fortunately good weather prevailed, so that the whole crew could put their entire energies on the hand pumps. All sail was set, but had no effect, the wind being perfectly calm. The ship was then about 176 miles from Queenstown. The speed at the time of the accident was about 19½ to 20 knots. On the morning of the 26th the first officer put out in the steamer route in the hope of finding a vessel to take them in tow. On the morning of the 28th he met the *Aldersgate* (2,000 I.H.P.), which towed them to Queenstown, where they arrived at 4 A. M. on the 30th; *en route* they met the *Ohio*, which also helped to tow them. During the whole time the hand pumps were worked to their full capacity, though all attempts to pump out the engine rooms failed. At Queenstown divers succeeded in closing the sea-valve openings on the starboard side, the water was pumped out, and on the 2d of April the vessel proceeded to Liverpool under her own steam with the port engine, arriving on the 3d. She remained there from the 4th to the 8th under the 100-ton crane for the purpose of removing the heaviest parts of the machinery, such as cylinders, covers, pistons and frames. On the 9th she was put in drydock No. 3, at Birkenhead, and on the 10th the hull of the vessel could be inspected.

Damage was found that could not have been imagined, and other places which were believed to be damaged were found perfectly sound.

The skin of the ship was intact, but the starboard screw, which

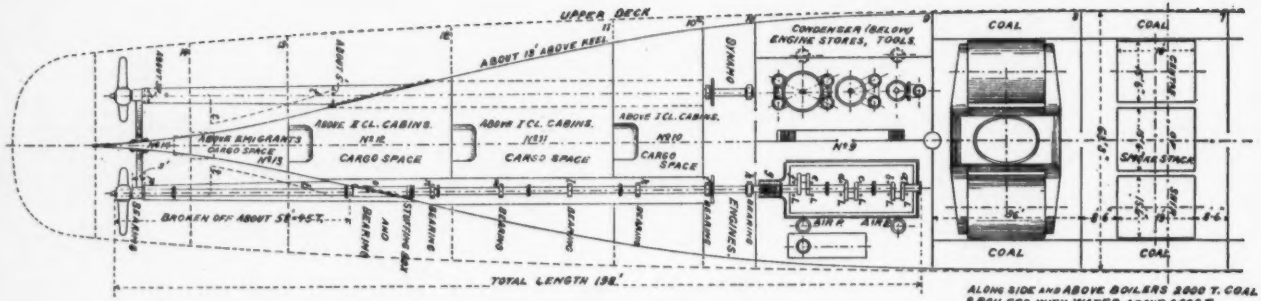
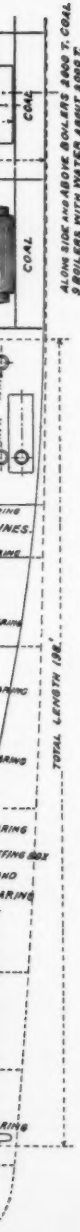

UPPER DECK
ABOUT 15' ABOVE KEEL
CONDENSER (BELOW) ENGINE STORES, TOOLS.
COAL
COAL
ABOVE EMIGRANTS CARGO SPACE No 13
ABOVE 2 CL. CABINS. No 12
CARGO SPACE
ABOVE 1 CL. CABINS. No 11
CARGO SPACE
ABOVE 1 CL. CABINS. No 10
CARGO SPACE
No 9
No 8
No 7
No 6
No 5
No 4
No 3
No 2
No 1
SMOKE STACK
BROKEN OFF ABOUT 52'-9" S.T.
STEERING ARM AND BEARING
BEARING
BEARING
BEARING
BEARING
ENGINES
AIR PUMPS
COAL
COAL
TOTAL LENGTH 192'
ALONG SIDE and ABOVE BOILERS 2000 T. COAL
9 BOILERS WITH WATER ABOUT 2000 T.
TOGETHER 4000 T.

FIG. 1.

FIG. 3.


COAL
COAL
ALONG SIDE AND ABOVE BOILERS 2800 T. COAL
3 BOILERS WITH WATER ABOUT 2000 T.
TOGETHER 4800 T.
TOTAL LENGTH 138'

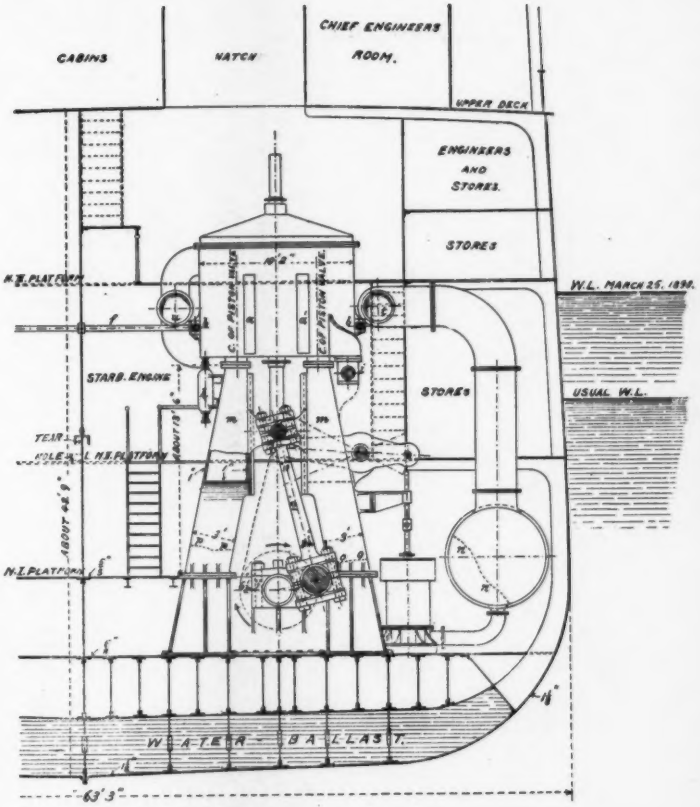
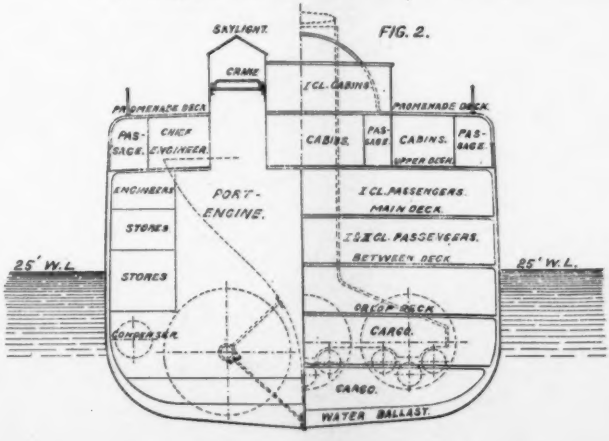

CABINS
HATCH
CHIEF ENGINEERS ROOM.
UPPER DECK
ENGINEERS AND STORES.
STORES
W.L. MARCH 25. 1894.
USUAL W.L.
STARB. ENGINE
STORES
WATER BALLAST.
63'3"

FIG. 2.


SKYLIGHT.
CRANE
I CL. CABINS.
PROMENADE DECK.
PROMENADE DECK.
PAS-SAGE.
CHIEF ENGINEER.
CABINS.
PAS-SAGE.
CABINS.
PAS-SAGE.
UPPER DECK.
ENGINEERS
PORT-ENGINE.
I CL. PASSENGERS. MAIN DECK.
STORES
I & II CL. PASSENGERS. BETWEEN DECK.
25' W.L.
STORES
25' W.L.
ORLOP DECK.
COMPRESSOR
CARGO.
CARGO.
WATER BALLAST.

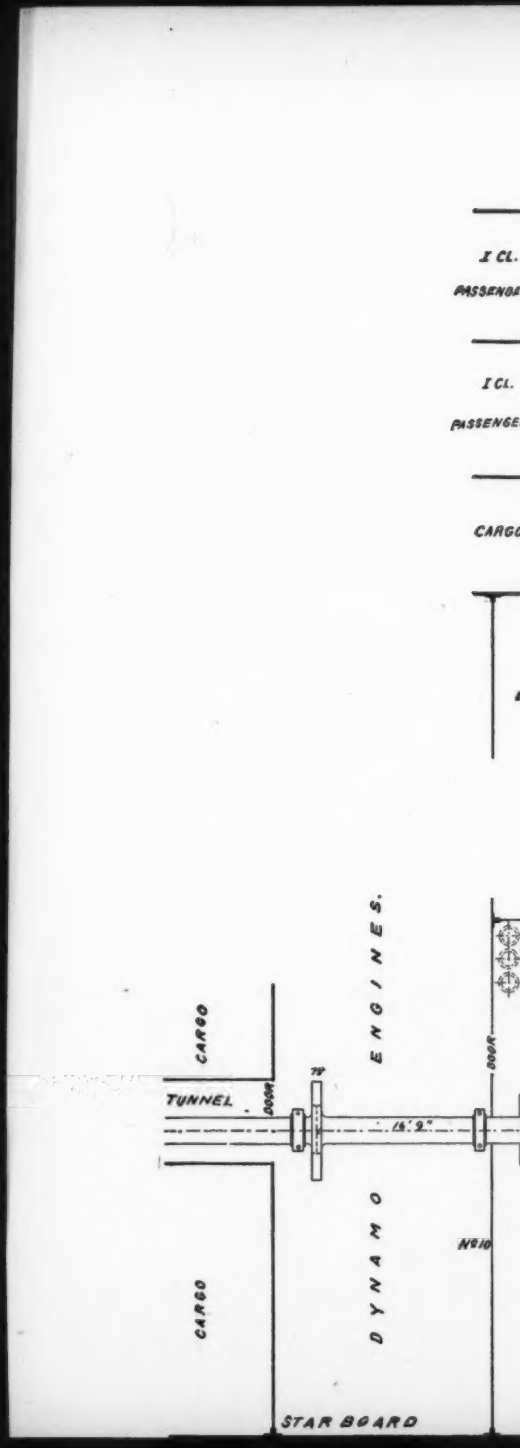

I CL.
PASSENGE
I CL.
PASSENGE
CARGO
ENGINES.
CARGO
TUNNEL
DOOR
DOOR
DYNAMO
CARGO
STAR BOARD

FIG. 4.

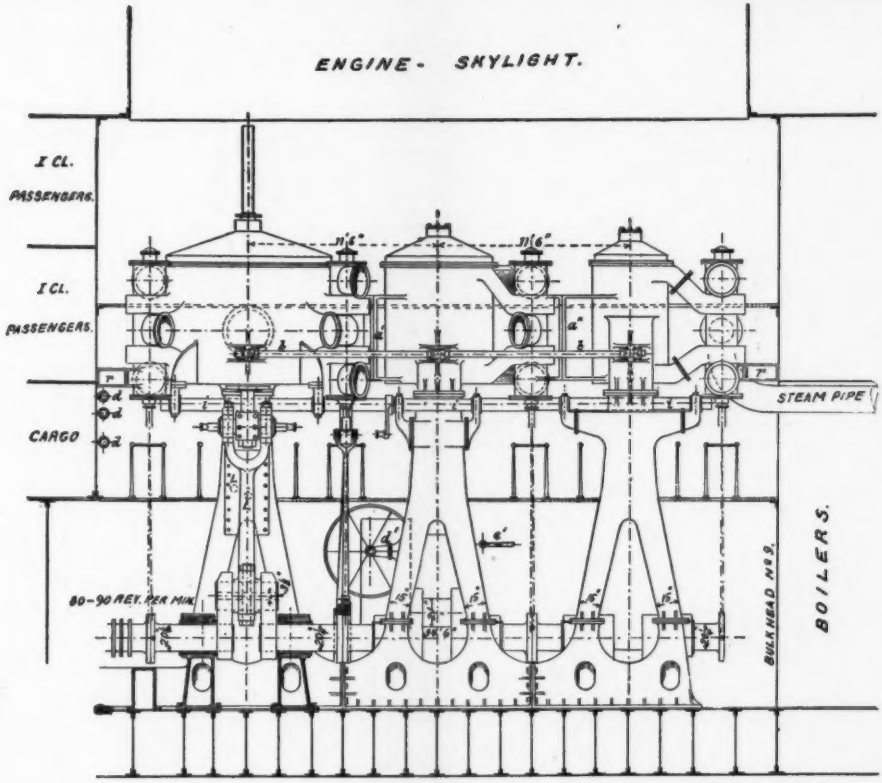
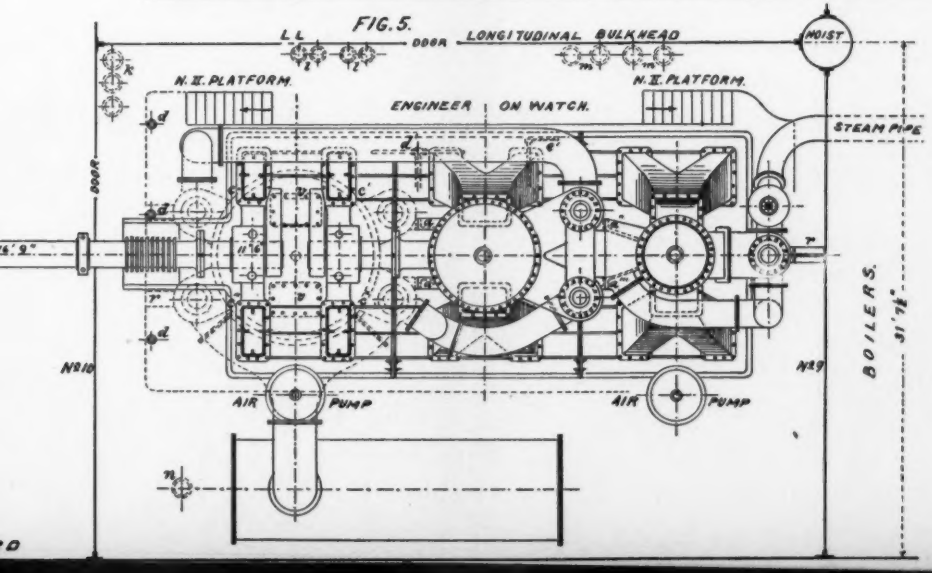

ENGINE - SKYLIGHT.
I CL. PASSENGERS.
I CL. PASSENGERS.
CARGO
STEAM PIPE
80-90 REV. PER MIN.
BULK HEAD No 9.
BOILERS.

FIG. 5.


LL
DOOR
LONGITUDINAL BULKHEAD
MOIST
N. H. PLATFORM.
ENGINEER ON WATCH.
N. H. PLATFORM.
STEAM PIPE
DOOR
No 10
AIR PUMP
AIR PUMP
No 9
BOILERS.

should have been hard up against the outboard bearing, was about 18 inches from it, and the shaft casing was buckled on the lower half of the forward and after ends, and partially worn through. Upon closer inspection it was seen that the shaft had dropped down about 8 inches from its normal position, that the lignum vitæ had disappeared, and that the brass lining which had contained the lignum vitæ was entirely worn through on the lower half, the shaft having ground itself deeply into the steel boss of the bearing. These facts explained everything: the shaft must have broken within the large casing outside the ship, and *this* was the cause of the racing of the engine on the 25th of March.

To find the break in the shaft, the rivets were cut from the casing next to the ship, but without taking any precaution in case the broken shaft should fall. The workmen had quit work, when about midnight Walls was informed that the shaft, with the screw, casing, bearing, etc., representing a weight of 46 tons, had fallen into the dock.

But what was the cause of this extraordinary wear on the bearing, while that of the port shaft was in perfect order? This mystery has up to this time not been solved.

V.

All possible theories have been advanced in regard to it.

a. The length of unsupported shafting was too great. From the stern tube to the outboard bearing there was an unsupported length of shafting of 47 feet 6 inches, or a free length of 28 diameters. This is evidently great. Large war vessels with engines of the same power as the *City of Paris*, *e. g.*, the French protected cruiser *Dupuy de Lôme*, have two bearings in order to reduce the length between supports. And it is only in smaller war vessels that the length of such shafting is greater. With similar large twin-screw steamers recently built, the unsupported shafting is in each case less; the proportion of diameter to length being as 1 to 20. On the *Augusta Victoria* of the Hamburg American Packet Co., the unsupported length is about 32 feet 8

inches and the diameter 20½ inches, a ratio of about 1 to 19, as against 20.5 to 570 inches on the *City of Paris*, or 1 to 28. About the same proportions exist on the twin-screw steamers *Columbia* and *Normannia*, of the same line. On the *Columbia* the length of shafting is 33 feet 6 inches; the shafts are hollow, 9 inches inside diameter and 21 inches outside diameter, the ratio, therefore, being 1 to 19. Objection might be raised to the argument against so long an unsupported shaft, that the bearing on the port side remained uninjured although it was working under exactly the same conditions as the one on the starboard side. Furthermore, both bearings of the *City of New York*, which had been in service eight months longer, are of precisely the same construction and are good to-day. But against this must be considered the possibility that the broken shaft of the *City of Paris* might have been badly lined up from the very beginning, that the screw struck something, etc., all of which can not now be proved.

b. The pressure on the lignum vitæ bearing was too great. Calculation, neglecting grooves and buoyancy, gives a pressure of 46.8 pounds per square inch. Seaton (Manual of Marine Engineering, page 175, the standard English work on marine engines) gives 50 pounds per square inch as the limit. The same value is given by Busley, Schiffsmaschine, Vol. II, page 101. According to this, the pressure was great though not excessive.

A comparison of these proportions on other twin-screw steamers gives:

Name of Vessel.	Weight of screw and half of outboard shafting.	Bearing surface.	Pressure per sq. in. of bearing surface.
Columbia.....	25 tons.	6 feet 3⅜ inches long, 22¾ inches diameter = 1,714.78 square inches.	32.65 lbs.
Normannia.....	30 tons.	5 feet 9½ inches long, 23¼ inches diameter = 1,615.87 square inches.	41.50 lbs.
City of Paris.....	35 tons.	6 feet long, 22½ inches diameter = 1,620 square inches.	48.40 lbs.

c. The lignum vitæ had swelled too much. An expert, Mr. P. Sampson, tried to prove this on the strength of experiments made by himself, as if John Penn, who 35 years ago introduced lignum vitæ as the most suitable material for the stern-tube bearings, had not himself made very accurate experiments. According to the statement of foreman Heron, only the very best material was used, and carefully put in during the repairs in January. The name Laird Bros., is a sufficient guarantee that the lignum vitæ was as good as could be had.

d. The lignum vitæ strips were not properly put in. (See Fig. 8.) This method of fitting the lignum vitæ strips has often been used before with good results.

e. The metal sleeve on the shaft was cracked, and the sharp edges along the crack, acting like a knife, cut away the wood, etc. The thickness of the sleeve which still remained on the shaft I found to be about $\frac{13}{16}$ inch. When new it must have been at least an inch, which appears to be none too thick. It sometimes happens that this sleeve cracks after having been worn down to a certain thickness, caused by internal stress in the material, due to rapid or irregular cooling off. The Inman Line had such a case happen four times in six years, with twenty-two vessels, though it never went further than to destroy the lignum vitæ; the entire bearing was never destroyed, nor the shaft broken. It does not, of course, follow that the destruction may eventually go farther.

f. Ashes got into the bearing and caused a grinding process. The official record of this is as follows: She burned American coal on the voyage, and there were more ashes thrown overboard on the starboard than on the port side (Walls' statement). But how could ashes get into the bearing? On this the Court does not offer the slightest clue. In Liverpool I heard that a big hole had been cut in the top of the casing, so as to allow the water to get to the bearing. This was corroborated in London. I did not notice this hole while inspecting the ship in drydock on May 2, partly because I gave my entire attention to the broken shaft, and having no knowledge of this hole, I did not look for it. Professor Busley, who had seen the vessel the day before, told me that he had seen where a plate about 12 by 16 inches had

been bolted to the forward end of the casing, with which this hole was probably closed.

Be that as it may, I know of some twin screw vessels that have no casing at all, and where the shaft runs naked in the water, *e. g.*, the two nearly-new duplicate steamers of the Hamburg American Packet Co., the *Columbia* and the *Augusta Victoria*, where the unsupported length of shafting is 33 feet 6 inches and 32 feet 8 inches, respectively.

Except for the counter arguments already made on the various points, all the theories from *b* to *f*, as was already done under *a*, can be reduced to a comparison of the port bearing of the *City of Paris* and both bearings of the *City of New York*, which, of exactly the same design and under similar conditions, are well preserved.

There still remains the theory that sand got into the bearing in some shallow place, and on account of the short time necessary to wear off lignum vitæ, it soon made the beveled edges disappear, thus causing a lack of opening for water, and consequently further destruction was rapidly augmented. But this method of fitting lignum vitæ having been used for years it is incredible that such an accident should not have long ago happened to some other vessel, there being many others that more frequently get into sandy and shallow places; and that this should have just happened for the first time, and to the *City of Paris*, is remarkable.

Equally fruitless was the examination of the court in regard to the destruction of the engine, viz: which part broke first, and how this rapid and complete destruction took place.

VI.

All these questions will be treated in the report of my trip to Liverpool. Arriving there at noon, May 2, 1890, I found myself at 5 o'clock, thanks to the advice of Mr. Busley, in Laird's boiler shop (Birkenhead); in the erection shop I found what was left of the L. P. cylinder, the cover and head, and the piston and columns. At first sight I was struck by the numerous

blow holes in the cracked valve chests, and the broken portions of the oval walls showed a thickness varying from $\frac{3}{4}$ to $1\frac{3}{4}$ inch. I could hardly believe my eyes. And the broken portions of the cylinder and liner were just as irregular; in one place it measured $1\frac{9}{16}$ and in another $2\frac{1}{8}$ inches. The court having called the attention of James Thomson to the varying thickness of the cylinder he admitted the fact, but could not explain the cause. The blow holes in the castings were not considered at all by the court. Besides, it was no easy matter to measure the exact thickness, as the broken pieces were very much battered.

The two broken parts of the cylinder (cc''', and $c'\,c''$ Fig. 5) together represent a little more than half of the circumference of the cylinder. The remainder was broken into a number of pieces of all sizes. The bottom of the cylinder was entirely knocked out. The thickness of the latter and of the ribs, as well as the number of the ribs, was, in my opinion, carried too far. The flange of the cover was still on the studs of the cylinder in the shape of small rings. In general I found all flange connections intact, while everything else of the same part would be broken to pieces, as, for example, the flanges on the columns, the one below being on the bedplate and the one above on the cylinder.

The latter was little injured in the upper part. All piston valves were in their chests. Only one of the four balance piston covers was broken off; the lower end of the valve rod, $5\frac{1}{4}$ inches diameter, was bent and the stuffing box broken; consequently the cylinder fell upon the lower end of this rod, and the balance piston, being driven upward, knocked off the cover. On one of the large quadrant-shaped brackets, to which a reversing shaft was secured (see Figs. 3 and 4), the bottom flange was still in place, though the bearing itself was broken square off. The line of rupture of the cylinder was nearly parallel to the axis, as might have been expected. The flat parts of the ports were braced by screw stay bolts $1\frac{3}{4}$ to 2 inches diameter with a very fine pitch, instead of cast ribs or smooth stays. These, as well as most bolts, were mutilated. The 24-inch copper steam pipes were still in place, but were very much flattened.

The cylinder cover was in a sad condition. The flange was entirely gone. There was a very prominent depression near the center, half an inch deep, but no trace of a crack, although the material, as well as that of the cylinder, was "short." I found the tail-rod stuffing box without the usual ring at the bottom.

Of the piston, I found about nineteen large pieces, the largest about 3 feet 3 inches. Close by were a large number of small pieces, some of the size of the hand. All pieces are remarkable from the fact that not one was deformed, but retained the original conical shape, while there were many depressions and flat pressed surfaces, a square foot or more in area, on the cast-steel columns; and all the edges of the broken surfaces were perfectly sharp, not distorted, and therefore coarse grained and crystalline. I tested the material with a sharp chisel, but could not make the chip roll up much; it always broke off at a length of about $\frac{3}{8}$ inch. Thinking that this might be due to my unsteady hand, not having done any chipping for over 40 years, I got a boiler maker to repeat the test. He tried it with a very sharp chisel, but he, too, could not cut a chip longer than $\frac{3}{8}$ inch. From my experience, I must conclude that this material was very brittle and lacked toughness. In contrast with this, the warped annular follower was of excellent material. The bolts for securing it were tapped into the body of the piston. The cast-iron piston packing rings were present in innumerable pieces; of the springs (Peck's piston packing was used on all the pistons) between the packing rings I found no traces. The thickness of the piston was somewhat less than the drawing, which I subsequently received in the office of the Inman line, called for. This also agrees with the calculation. According to the drawing (Fig. 6) the piston should weigh 5.96 tons, but it only weighed 5.5 tons (Thomson's statement).

The fragments of the columns were nearly all in heaps, making an examination very difficult. The largest pieces did not measure more than 3 feet 3 inches by 4 feet 3 inches. The material was cast steel, and appeared to be very brittle. The chisel tests gave only a little better result than that of the piston. A piece a foot square was cut out of one piece for the purpose, as I was informed, of making tests for the Board of Trade; but

during the entire proceedings of the Court not a single word was said about it, nor of the material of the piston, to which I had specially called the attention of the Inman people.

The calculated weight of these columns, according to the drawings, was 6 to 7 tons, with an average thickness of 1 inch, and the flanges $1\frac{3}{4}$ inches. As a matter of fact, they weighed 7 tons, according to Thomson's statement. Instead of an average of 1 inch, it actually varied from $\frac{5}{8}$ to $1\frac{3}{4}$. It is apparent that such a steel casting must be full of stresses, and it was acknowledged that five of the six columns of the starboard engine were patched. These patches consisted of steel plates fitted on while hot and secured by a number of tap bolts. According to the statement of an expert, the fractures occurred all over, but not where the patches were. I did not see any, nor did I look for them, having learned of their existence afterwards. The holes for the $3\frac{1}{4}$-inch bolts which connect the piston with the piston rod flange showed that the piston was separated from the rod by a pressure from below. This could not be otherwise, because all the bolts and nuts were found on the piston rod flange. I refer to all these circumstances because they serve to explain the true proceedings during the destruction.

I thought I had seen the most incredible things, and was about to withdraw, it being after working hours, when the guide asked me if I had looked around the yard, and then showed me the way to go. In six different heaps, each about 20 by 33 feet, some as high as three feet, were an innumerable quantity of fragments of the columns (also the two planed crosshead guides, about 2 inches thick, which singularly enough were in large square plates), cylinder liner, etc., eccentric rods, links, reversing and air-pump shafts; the latter with air-pump beam of $\frac{1}{2}$-inch plate, showed but few broken pieces, but was bent and twisted in such a way that it would have been impossible to have made a drawing of it. The air-pump beam shaft, about 10 inches diameter, was bent to a semi-circle, and the ends terribly crushed. Of the air pump the foot-valve seat, broken in three pieces, and the hot well (of brass) were the only recognizable remains to be found. Of the many valves, 6 inches in diameter, not a single

one could be turned; they with their guides, were so badly crushed, broken and warped that it was impossible to tell whether they were rubber or metal valves.

The heads of the large 3¼-inch bolts of the outboard and propeller-shaft flanges were ground off on one side down to the body of the bolts, and several of the nuts, 6 inches deep, were ground down to a thickness of ¼ inch. (This occurred on the voyage to Queenstown and from there to Liverpool, as the broken shaft with the flange ground through the casing, the screw being able to turn freely on the voyage). The metal lining, about 6 feet long, in which the lignum vitæ was fitted, showed an opening below about 20 inches wide, as if it had been planed out. The thickness of the liner was 1 inch. The crack on the steel boss of the strut bearing showed a fine grain; as also the two steel caps of the crank-shaft bearings, which were broken square in two. A separate large pile was formed by the large copper exhaust pipes, about 40 inches diameter, and ¼ inch thick; they were pressed oval, rolled together like paper, and full of holes. The portions of platforms, ladders, hand-rails, supports, horse shoes of thrust bearing, levers and handles, oil boxes, drip pans, the six cast-iron traps, etc., were hardly recognizable as such in the heap. I found some pieces connected that did not belong together, as for example a ¾-inch bolt mashed perfectly flat around the broken surface of a piece of cylinder liner, apparently as if welded to it. The two 3½-inch bolts, about 6 feet long, for securing the horse shoes on the thrust bearing presented a peculiar appearance; they were twisted in fantastic shape, but there was not a nut missing. A Tangye engine with brass centrifugal pump, apparently belonging to the auxiliary condenser for the steam accumulator pump, was perhaps worth repairing.

Close to Laird's boiler shop is the old Mersey forge, which I had seen 35 years ago; it looked to-day as it did then, but appeared rather small and obsolete. A number of large broken propeller and paddle-wheel shafts were lying near the entrance. I was examining them with great interest when the foreman came up to me, and at once commenced to speak about the

FIG. 6.

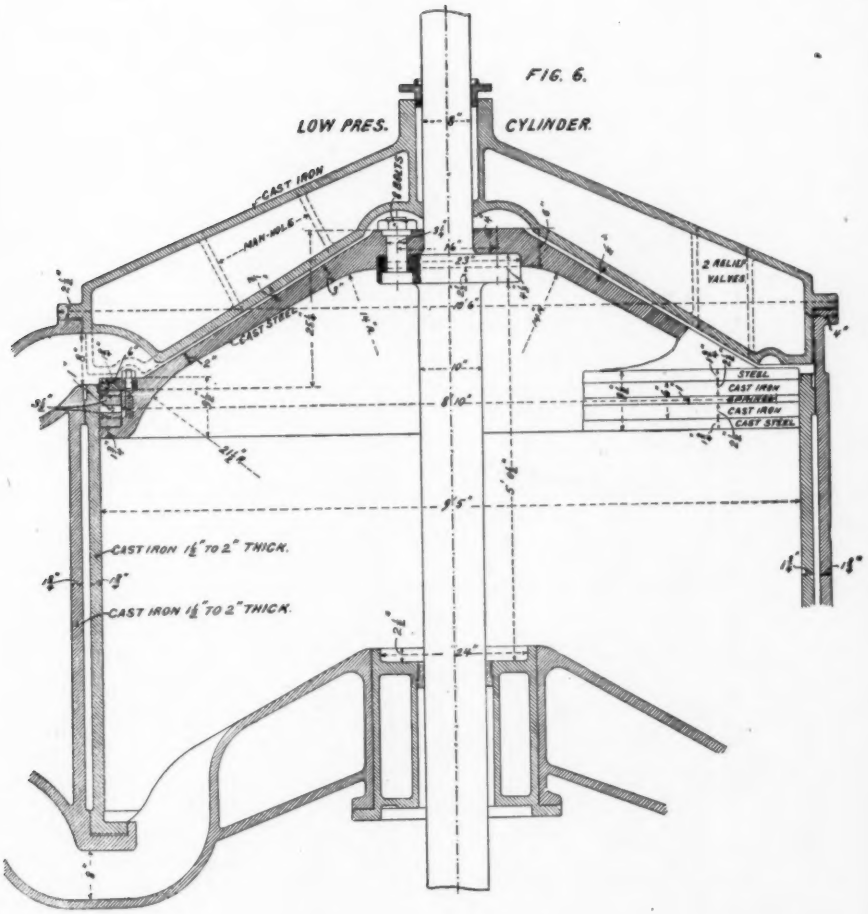

LOW PRES.
CYLINDER.
CAST IRON
MAN-HOLE
CAST STEEL
2 RELIEF VALVES
STEEL
CAST IRON
CAST IRON
CAST STEEL
CAST IRON 1½" TO 2" THICK.
CAST IRON 1½" TO 2" THICK.

FIG. 7.

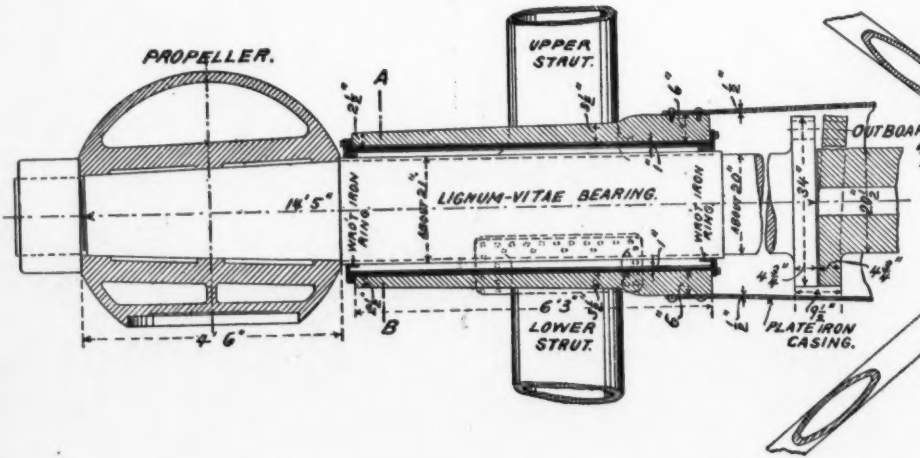
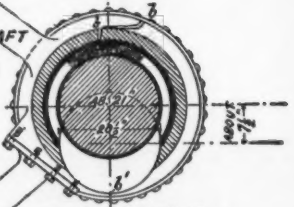


FIG. 8.



SECTION ON A-B.

FIG. 9.

accident to the *City of Paris.* He said: "The people are puzzling their brain to know how the accident happened, but just look at that stuff in the cylinder—it *had* to break." I mentioned this merely because this accidental testimony of an unconcerned party was of value to me. Though I do not want to assert that a better, or the very best material could have prevented the accident, it is nevertheless strange that the experts on the court maintained that the material of hull, engines and everything was good throughout.

I found the *City of Paris* in dock, and as it was beginning to get dark, I made all haste to get down into the dock so as to have a look first of all at the broken shaft and the construction of the strut bearing, though particularly at the shaft fractures, before the parts were removed. On the propeller shaft were two brass rings, about 12 inches long, the remains of the brass lining, and they were loose. The fractures were unusually coarse grained, though from this the quality of the material cannot be judged, as the shaft and the lining no doubt got very hot from friction. The wrought-iron rings, which held the ends of the lignum vitæ together, were still on the shaft. They were considerably worn off, about $\frac{1}{2}$ inch, where they had been rubbing on the bearing, but otherwise were not damaged. The coupling flange of the hollow outboard shaft was broken (see Figs. 7 and 9). The fracture was scaly, exhibiting regular loose scales, and was entirely different from the fracture of the shaft in the stern tube, a piece of which, about 18 inches long, was still in place. This fracture was almost at right angles to the axis, and very smooth, but not ground off. The fracture in the steel boss on the strut bearing formed above an angle, and below was parallel with the axis. The latter fracture was very short, as the boss for three-fourths its length was entirely worn through. The blades of the screw propeller seemed to be of very good material. One of them was rolled up at the tip, another was badly damaged. This happened on the night of April 11, when they fell into the dock. There were a number of zinc plates on the bottom of the ship to prevent galvanic action, yet the nuts which secured the blades to the hub were honey combed, though only one year old.

I looked at the damage to the bulkheads and the engine room hatch, the greatest of which was on a line with the promenade deck, about 40 feet above the center of the shaft, immediately below the traveling crane. The ¼-inch plate was stove in for a space of about four by six feet. The traveling crane was likewise bent in several places. The engine-room bulkheads were also damaged in many places.

Above the crank shaft, on a block and fall in the hatch, hung the connecting rod with crosshead and piston rod, but so high up that I could not examine the damages individually (two on the upper and four on the lower end). All the connecting-rod bolts were in their places. The rods were bent at the upper end, as reported in "Engineering," but in the center, and about 12 inches towards the port side. On the crosshead pins only a few splinters of the guides were noticeable. The piston rod was bent to an angle of 90 degrees. The eight bolts which connected the piston to the flange of the rod were still in place, together with the nuts and a few remnants of the piston. The nuts were pressed entirely out of shape, and the flange was split in several places. On the circumference whole segments were missing. (This flange and the connecting rod did the main work of destruction. The bed plate could not be seen, it being still covered by fragments and foot planks.) Bearings and shaft were well exposed. One of the bolts of the after crank-shaft bearing was broken off deep in the bearing: the others were broken square off close to the bed plate. One of the bolts of the forward low-pressure crank-shaft bearing was very badly bent, both bearing caps were broken, and the bed plate showed many large cracks, especially under the bearings.*

The shaft was very much bent. The flange connections on the feet of the columns were in good condition. The greatest height of what was left of the columns above the bed plate on the star-

*One of the bed plates of the *City of New York* was being patched at this time. The crack extended from the bearing down about 3 feet and opened ⅛ inch at the bottom. In the vicinity of the crack on the lower flange 3 holding-down bolts were shorn off. It was, therefore, a crack due to internal stress.

board side, was 16 inches, and on the port side about 24. Crank pins and brasses were in good order, except that the cranks were twisted $\frac{1}{8}$ inch forward on the shaft. Only the lower part of the thrust bearing was left, and that was badly damaged. The condenser presented a terrible appearance. The part *nn* (Fig. 3) for a length of about 5 feet, was smashed off, the tubes flattened to the thickness of a table knife. Besides the holes already mentioned, the fore-and-aft bulkhead showed many other damaged places, *e. g.*, a zig-zag rent behind the Z-iron stiffener and nearly covered by it. This rent was inaccessible, and even if it had been possible (after the accident) to close up the two 12-inch holes, the water would still have found its way through this rent. The water rushed into the starboard engine room immediately after the condenser was damaged, not only through this and the 15 and 24-inch sea connections of the circulating pump (both of which were, according to the Admiralty requirements, below the water line) and other sea connections, but also in three other places below the engine foundation.

Beneath the low-pressure crank is a $\frac{5}{8}$-inch plate, covering the top of the foundation, and forming the bottom of the crank pit, the bed plate resting upon it. There was a hole in this plate about 12 inches diameter, caused no doubt by a piece getting beneath the crank shaft, and being driven below, piercing the upper plates of the double bottom. The outboard openings being from 16 to 21 feet below the water line, the engine room must have filled very rapidly. The flange *a'* (Fig. 3), 2 by 8 inches, was entirely missing on the I.P. cylinder, even the 2-inch connecting rib being broken off. Besides this, several cast facings on the columns were missing, the reversing shaft and hand-gear brackets were broken off, as also the lugs *b* (Fig. 3) for the side tie rods, etc. That all pipe connections between the I.P. and the L.P. cylinders were broken is self evident.

The total damage estimated by the Inman Line is about £100,000. For towage alone, the Admiralty Court of London allowed the *Aldersgate* £7,500 and the *Ohio* £600. The original cost of the *City of Paris* was over £350,000.

VII.

A short criticism of the engines may now be in place. As regards design, the piston valves of the L.P. cylinder have already been mentioned under II. If slide valves had been used, as most designers rightly do for the L.P. cylinder, the clearance would have been about 9 instead of 15 per cent., and the compression would have been twice as great. In using the ordinary link with two eccentrics the engines occupy about 5 feet 6 inches more space than would be required with Joy or other similar gear with one eccentric. By putting the eccentrics on the shaft couplings instead of on the shaft, they have to be made about 15 inches larger in diameter than in the latter case. Therefore, an appreciable amount of work is lost by friction on the six eccentrics.

Figs. 13 to 15 are indicator diagrams taken on May 5, 1889, during a fast voyage, with 19,900 I.H.P. The combined diagram, Fig. 16, shows no deviation from other triple-expansion engines, as, for example, those of the *Meteor* of 3,000 I.H.P., tried by an English commission.

The exhaust of the auxiliary machinery, amounting to 250 I.H.P., is hardly appreciable in the upper line of the L.P. diagram. (With this arrangement the pressure of the exhaust steam of the auxiliary machinery is about 12 pounds above the atmosphere, and all exhaust pipes are to be considered as receiver spaces of the L.P. cylinder.) The clearance assumed in Fig. 16 will differ little from the actual. Mr. Thom, the inventor of the much-used compression slide valve, whom I met in the office of the Inman Line, estimates the lower clearance space of the L.P. cylinder at 20 per cent. Under such conditions it is impossible to get the necessary compression for the smooth running of an engine. The distribution of the steam with four piston valves is therefore in no case to be recommended as an improvement. As the diagrams show, the steam follows very far—.75 in the L.P.—and the compression hardly averages 11 per cent. (Upon inquiring if an earlier cut-off had been tried, so as to get greater compression, chief engineer Walls answered that the greatest

power was obtained with a late cut-off, and that he didn't care anything about compression.)

The indicator diagram shows what takes place inside the cylinder, but is far from giving a picture of the actual pressures transmitted to the crank pin. To construct these diagrams, *e. g.* Fig. 15*a*, first draw in on the diagram in question the lower line of the opposite end of the cylinder, and then set off from a horizontal line the diagram thus obtained, so that it assumes the shape *abcde*, in which, under certain conditions, the part *de* falls below the line *ae*, and is therefore negative. The acceleration curve is then drawn so that the neutral line falls on *ae*. The upper part of the curve *aa'* cuts off the unshaded portion of the indicator diagram, while the lower part increases its area.

The average pressure of acceleration, *i. e.*, the line *ea* is equal to $\frac{0.00034\ W \times n^2 \times r}{F}$, in which W is the total weight of the rods, piston and crosshead, in pounds; n, the number of revolutions per minute; r, the radius of the crank shaft in feet; F, the piston area in square inches. The ordinates of the curve $a'a'$ (for connecting rods having a length of four cranks) are easily obtained from the auxiliary diagram in Rigg's Treatise on the Steam Engine, p. 274, which diagram can be used for connecting rods having a length of three cranks.

In these diagrams the horizontal-section lined surfaces represent positive, and the vertical negative work, *i. e.*, work such that the rods, together with piston and crosshead, must be moved by the rotating masses of the engine or by the other cylinders. By constructing the diagrams in this manner the upper line of the indicator diagram remains unchanged, so that one can see at a glance how the diagram of acceleration can be improved by increasing the initial pressure, by varying the point of cut off, by throttling, or by changing the exhaust lead and compression. (The usual method of drawing the acceleration-pressure diagram is to let the neutral line of the curve $a'a'$ be tangent to the highest point of the indicator diagram. The point here is totally lost, as for every change of the above factors a new diagram must be drawn. Furthermore, the drawing becomes very minute, and the result is a distorted and artificial diagram.

A glance at Figs. 13*a* to 15*a* shows that on the down stroke of all three cylinders the crank must cover the distance from the dead centers *a* to *K* before the steam pressure can overcome the pressure of acceleration; consequently one repeated change of pressure takes place in the connecting rod brasses, so great indeed that during one revolution the pressure on the brasses is changed three times, or nine times for all three crank pins The result is a variable thrusting motion under which all bearings and guide surfaces must suffer.

An improvement of the diagrams that gives a uniform and easy motion, could only be obtained by making extensive alterations, lightening the rods, giving greater compression, earlier cut-off, and change of crank-pin velocity. All diagrams are drawn for 89 revolutions per minute. Diagrams with a lower crank-pin velocity (of late the engines have been making from 82 to 84 revolutions) show no improvement over 13*a* to 15*a*, because if the curve *aa* descends, the initial pressure becomes less. I have no diagrams of the engine running without load (equivalent to engine racing). I tried to construct them on the assumption that the mean pressure would be about 15 per cent. of the normal performance. They are not given on account of the enormous pressure of acceleration at 180 revolutions, and could not be drawn within the limits of Figs. 13*a* to 15*a*, as an entirely different construction would have to be selected for them in order to give a clear idea of what takes place.

It may be observed that at 200 revolutions the terminal pressure of acceleration on the up stroke of the H.P. cylinder is 1,010,746 pounds, on the I.P., 1,167,540, and on the L.P. 1,446,-940. According to the latter there would be a stress on the bolts of the lower end of the connecting rod (about 33 sq. in. in section) of about 21,582 pounds per square inch, which almost reaches the elastic limit. As they did not break, it must be accepted that the destruction of the engine took place at less than 200 revolutions, the more so as in this condition not less than nine changes of pressure of great intensity took place during each revolution.

Figs. 13*b* to 15*b* represent the tangential pressures on the crank-

pin considering the length of the connecting rod, the pressure of acceleration, etc.

In Fig. 15*b* the outer dotted line shows the three tangential pressures combined. From this it will be seen that the pressure varies during each revolution from 100 (p) to 225 (P). And the fact that the weight of the rods is far from being balanced is not taken into account. (And here the change of pressure curves into perfect constant ones is remarkable compared with the wide variations of pressure in Figs. 13*a* to 15*a*, because the pressures at the ends of the latter are wholly within the bearing on the tangential diagram.) The nine changes of pressure for each revolution are shown at *K* in Fig. 15*b*.

These investigations lead further to this conclusion that the principle hitherto held of making the rods of all the cylinders of the same diameter is an assailable one. The principle of similar spare parts is, in my opinion, correct only for coupled engines having the same size cylinders; it is not even suitable for compound engines, because the principle of distributing the entire work of the engines equally between all cylinders is not possible, and it is entirely wrong for triple-expansion engines. To what condition this principle leads, can be seen on the *City of Paris*, where at the maximum performance of the engine (Figs. 13 to 15) the stresses on the 10-inch piston rods for the H.P., I.P., and L.P. cylinders were 1,390.6, 1,589.2 and 1,901.4 pounds per square inch, respectively. The average weight of the reciprocating parts of stationary engines with the necessary compression is about 4.0 pounds per square inch of piston area, while on the *City of Paris* it was 3.40 pounds on the L.P., and 7.09 on the I.P., while on the H.P. it was as high as 15.32 pounds.

The engine weights in tons, according to Thomson's statement, were for each engine:

	H.P.	I.P.	L.P.	
Piston,	1.15	2.45	5.50	
Piston rod, crosshead and slipper,	4.00	4.00	4.00	15.3
Connecting rod,	5.8	5.8	5.8	
Crank shaft,	14.00	14.00	14.00	*2.26
Bed plate,	16.00	16.00	16.00	
Columns (two),	14.00	14.00	14.00	

*Cranks alone for one cylinder.

The proportion of the cylinders cannot be called good. The best performance for the H. P. cylinder was 2,822, I.H.P., that of the I.P., 3,228, and that of the L.P., 3,900, or in the proportion of 72, 83 and 100. The voyages give results differing little from the following:

	August, 1889, 83 revs.	August, 1889.	Nov. 1889, 87 revs.
H.P. . .	2,440 = 68	2,523 = 72	2,688 = 74
I.P. . .	3,006 = 83	2,611 = 74	2,923 = 80
L.P. . .	3,595 = 100	3,512 = 100	3,648 = 100

In every case the L.P. cylinder developed about 1,000 I.H.P. more than the H.P.

The design of the steam jackets is a very imperfect one, in that they are carried only around the barrel of the cylinders, and not over the covers and bottom, and that the drainage depends entirely upon the engineer. This is of vital importance to the L.P. cylinder, for had the cover and bottom been jacketed the terminal pressure in compression would have been much higher, and the advantages resulting therefrom would have been obtained.

As regards workmanship, I found it externally, at least, very good. The H.P. valve chest was bolted to the cylinder, but the others were cast with the cylinders, and as far as I could learn—at any rate, on the L.P.—they were without valve liners. The valves have cast-iron packing and spring rings. I saw one of the valves of the port engine—the rings were black and showed little wear.

As regards material, it has already been mentioned under VI. The total weight of steel castings for both engines was 450 tons. All cast steel parts that came up for discussion were full of false stresses; this may also be assumed for the L.P. piston (Fig. 6), in which the distribution of material was very unequal. More than half of all the metal was near the hub and close to the circumference, and these parts must have cooled off much later than the conical body of the piston. The average thickness of

FIG. 13. H.P. CYL.

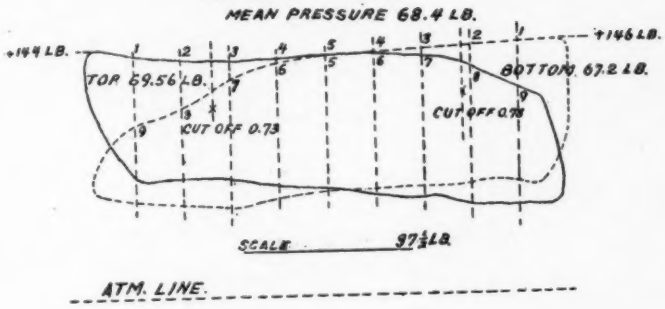


FIG. 14. I.P. CYL.

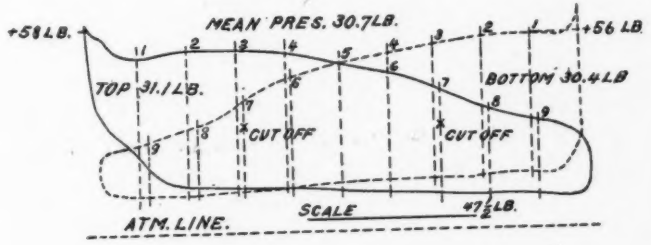


FIG. 15. L.P. CYL.

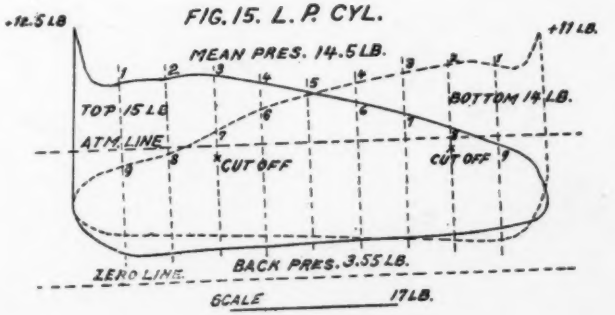


FIG. 13 Sq H.P. CYL.

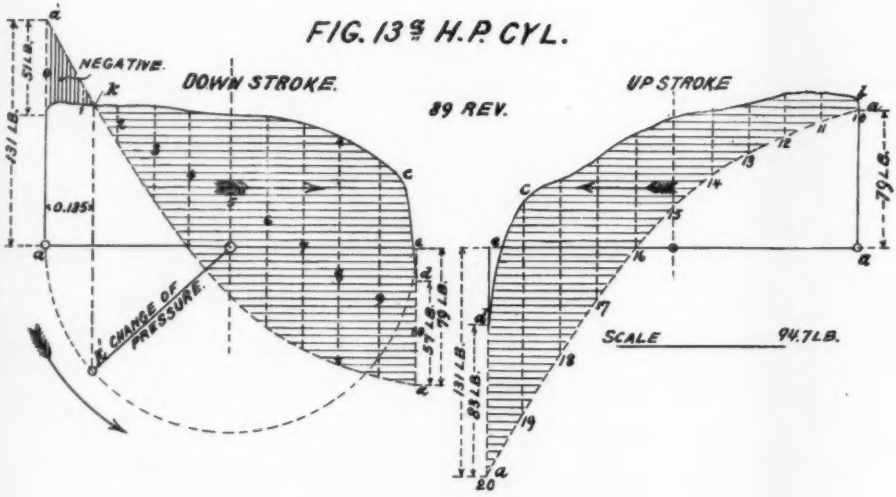
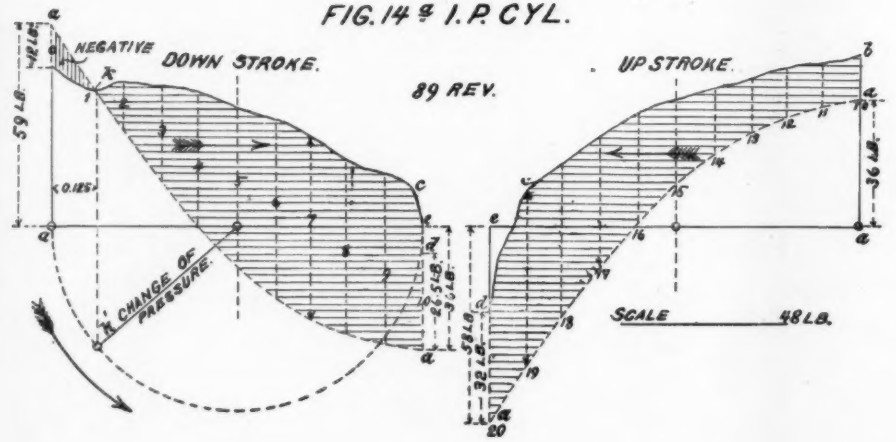


FIG. 14 Sq I.P. CYL.



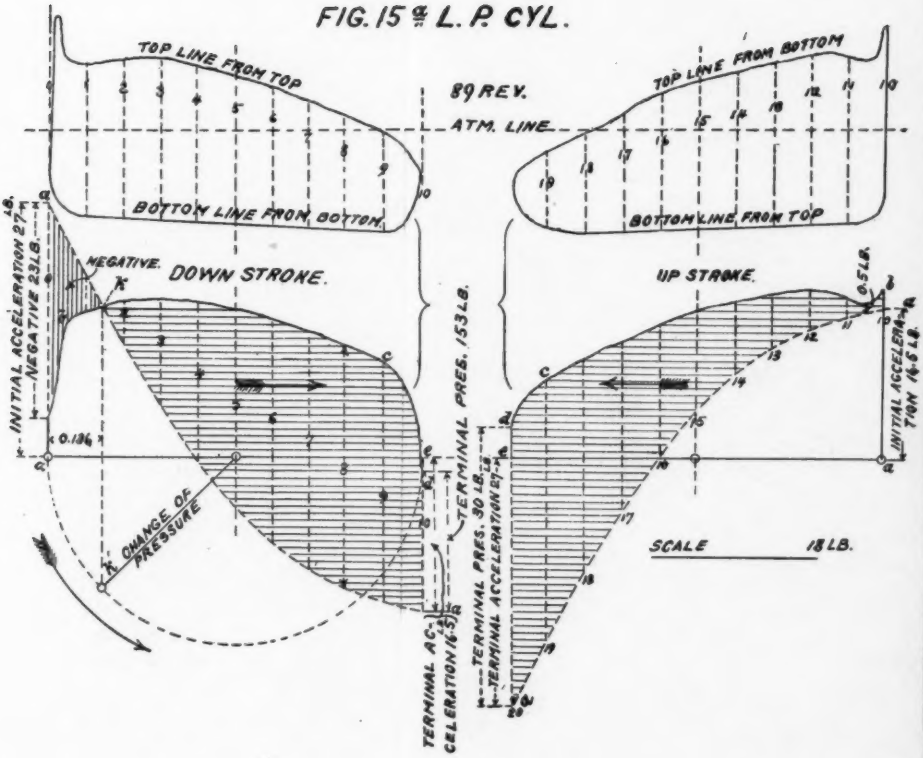


FIG. 15 of L. P. CYL.
TOP LINE FROM TOP
89 REV.
ATM. LINE
TOP LINE FROM BOTTOM
BOTTOM LINE FROM BOTTOM
BOTTOM LINE FROM TOP
INITIAL ACCELERATION 27 LB.
NEGATIVE 23 LB.
NEGATIVE.
DOWN STROKE.
UP STROKE.
0.5 LB.
INITIAL ACCELERATION 14.6 LB.
0.136
CHANGE OF PRESSURE
TERMINAL PRES. 153 LB.
TERMINAL ACCELERATION 6.5
TERMINAL PRES. 30 LB.
TERMINAL ACCELERATION 27
SCALE 18 LB.

FIG. 13½

H. P. CYL. (I).

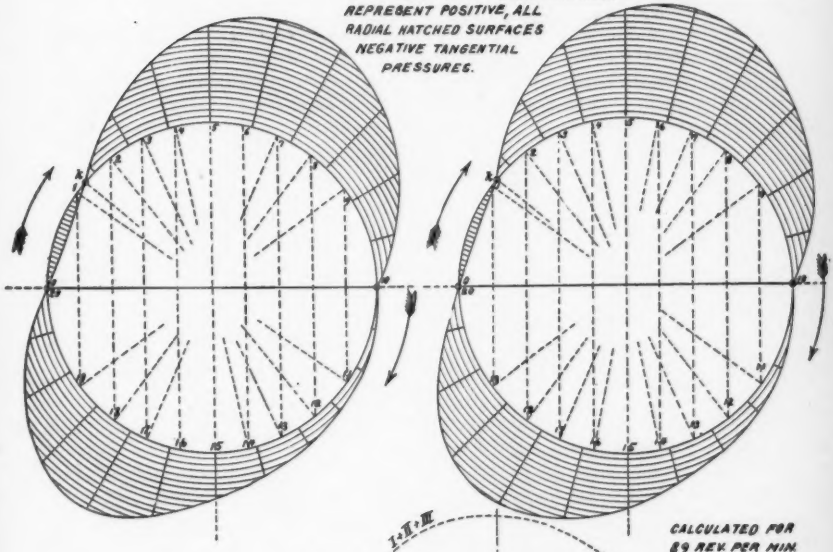
SCALE 12 LB.

FIG. 14½

I. P. CYL. (II).

SCALE 48.5 LB.

ALL CONCENTRIC HATCHED SURFACES REPRESENT POSITIVE, ALL RADIAL HATCHED SURFACES NEGATIVE TANGENTIAL PRESSURES.



CALCULATED FOR 89 REV. PER MIN.

FIG. 15½

L. P. CYL. (III).

SCALE 18 LB.

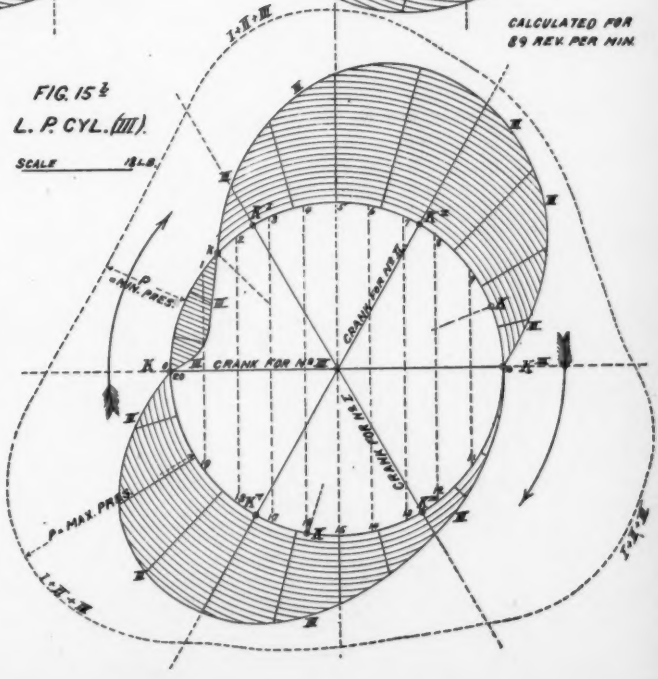
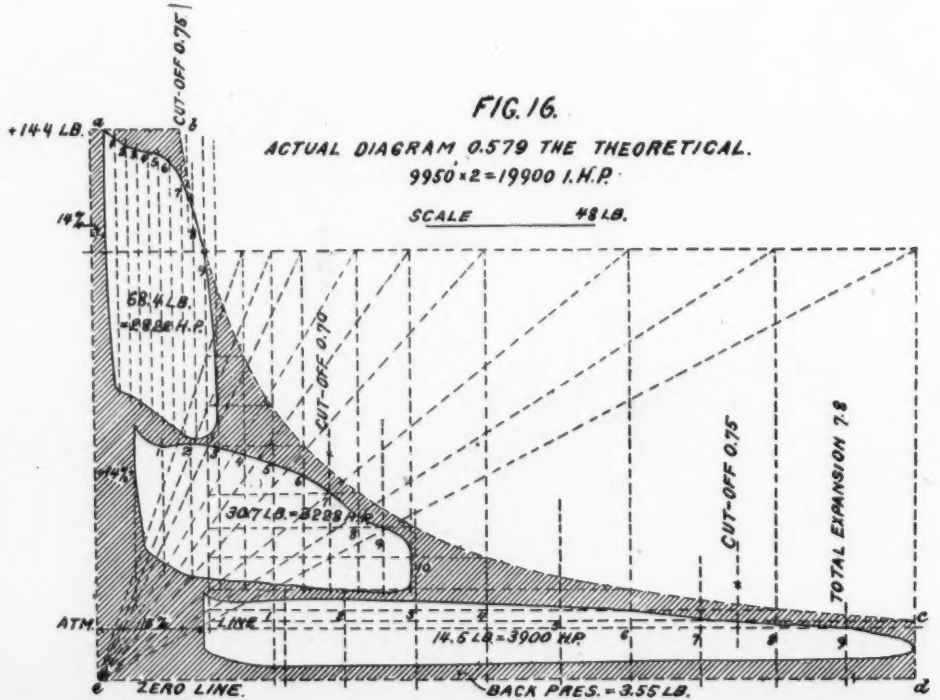


FIG. 16.

ACTUAL DIAGRAM 0.579 THE THEORETICAL.

9950 × 2 = 19900 I.H.P.

SCALE 48 LB.



the conical part was only about $\frac{1}{45}$ the diameter of the L.P. cylinder. For a number of pistons for the latest war vessels of the United States Navy this value is $\frac{1}{31}$; and on one of the latest vessels by Laird Brothers, $\frac{1}{34}$. Considering that during the racing of the engine the springs between the rings probably broke or were mashed flat, and that the cast-iron packing rings, weighing 1,650 pounds, acted like a hammer on the circumference of the piston, and in the same direction and at the same dead center as the moment of inertia of the mass of the piston, then, if the slightest amount of false stress was added to this, the piston had to break.

VIII.

We now come to the probable causes of the accident on March 25, 1890.

The cause of the racing was the breaking of the propeller shaft. Under V, I tried to show that the causes accepted by the experts of the Court in explanation of this break were insufficient. The true and very simple cause, in my opinion, is borne out by the following considerations:

During the night of April 11, when ~~both~~ the propellers fell into the dock, the coupling flange of the 42-foot section of outboard shafting was broken half off (see Figs. 7 and 9).

The casing from which the rivets had partly been removed broke first, being no longer able to support the weight of the shaft on the lower side. As the forward end of the shaft inclined aft during the fall, the large strut bearing broke (Figs. 7 and 8), and then the after end with propeller fell down. The height of the fall of the forward end of the shaft was about 16 feet, and that of the propeller, before the blades touched the bottom, 7 feet.

Now as the shafts tumbled into the dock—the total weight of both and of the propeller was 46 tons—the pressure exerted on the flange connection during the shock was an equal one; that is, each flange received an equal amount of the impact. If both flanges were equally strong (and they were) and sound, and if the force of impact did not reach the breaking strength of the

metal, then both flanges must have remained uninjured; in the other case they would have broken. But this was not the case. On the flange of the propeller shaft I found not the slightest fracture; it stood exactly at right angles to the shaft, at least as far as it was possible to judge with the eye, while the flange of the outboard shafting only, was completely broken, as shown in Figs. 8 and 9. This fracture could only have been the extension of a smaller fracture before the shaft fell, and no doubt dates back long before January 6, when the first appreciable wear of the lignum vitæ in the bearing was discovered. In consequence of this defective flange, the shaft could not revolve centrally in the bearing.

Several experts like Ramsay and Sir Reed acknowledged that the rapid wear of the bearing could only be explained upon the assumption that the shaft did not run true, but that this was not to be considered, without, however, endeavoring to prove its "impossibility."

The outboard shaft was a heavy and an exceptionally long piece. If the metal was poured in where the broken flange was, and more had to be poured in afterwards, an incomplete place could have easily formed, which growing weaker would eventually result in a fracture. At any rate, the weak place would have been noticed by fine flaws if the shaft had been taken out in January. And as the rapid wear of the bearing must have told every one that an unusual cause was active here, it was therefore thoughtlessness of all concerned not to look until it was found.

What a condition the bearing was in when the shaft broke, naturally can not now be ascertained. After the lignum vitæ was once gone, the shaft became very much inclined and the bearing surface in this state was considerably reduced, so that the pressure on the bearing was increased from 46.8 to 142 pounds per square inch, and had the voyage only lasted a few hours longer, then not only would the shaft have broken, but the cast-steel strut bearing would have gone to the bottom.

Here we see further carelessness. If a propeller shaft bearing is worn, say $\frac{1}{4}$ to $\frac{5}{16}$ inch, on a shaft of about 10 inches diameter, then every attentive engineer must notice it. The stern

tube shaft turns unevenly, runs warm, and requires a great amount of cooling water. These are such unmistakable signs that even during the proceedings of the Court surprise was expressed that nothing like this was noticed.

With a vessel with two screws, and being in a position to compare the motions of the two propeller shafts so as to detect possible differences, it is doubly unpardonable that not one of the nineteen engineers of the *City of Paris* noticed anything suspicious.

How did it happen that the L.P. engine, after the breaking of the main shaft, could have been so completely demolished in such a short time? The finding of the Court was that on account of the great speed of the engine while racing, the extension of the piston and connecting rods exceeded the ½-inch clearance between the piston and the cylinder cover, and that consequently the latter and the piston were demolished, and that the rest must have followed as a matter of course. Assuming a compression pressure of 34 tons—corresponding to the diagrams Figs. 13 and 15—and 15 tons for the total weight of the reciprocating parts, the engine would reach more than 500 revolutions per minute before the extension would amount to ½ inch, and as the compression during racing was not known (in any case it was not greater than shown in Fig. 15) this theory is out of the question.

That it could not have been excess of compression that was the cause of the cylinder cover being knocked out, I have already shown, and a further argument against this assumption can be added that the cover was not jacketed, and that the compression could not, therefore, have reached an excessive pressure.

Several articles in the "Engineer" and "Engineering" assume that on account of the moment of inertia of the piston, etc., on the up-stroke the bearing bolts broke first; but the two bolts in the lower end of the connecting rod would have to break first, as they had to resist the same pressure as the bearing bolts. All six bolts were 5½ inches in diameter; therefore, the connecting-rod bolts were subjected to twice the pressure per square inch that the bearing bolts were, yet they were uninjured.

Others look for the cause in the patched columns. It is impossible to see where the power could come from to destroy the two columns by compression or tension, unless a piston break were assumed. The cross section of these columns at the lower end, near the feet, was at least eleven times, and near the top, at the weakest point, at least six times as great as the lower (threaded) end of the piston rod. Another opinion was to the effect that during racing the boilers must have foamed, and that the destruction of the L.P. cylinder was due to water. Disregarding the fact that, on account of the large volume of steam in the main steam pipe, this water from the boiler could not have reached the engine during the few seconds of racing preceding its destruction, it is incomprehensible that the H.P. cylinder, which was also fitted with a piston valve, did not break first. A valve of this type is known to be dangerous in a case of this kind, because it cannot be raised from its seat like a slide valve, and the relief valves are, in such cases, entirely inadequate.

As a result of my investigation, based solely on facts, I have reached the following explanation:

As the engine began to race immediately after the inevitable break of the propeller shaft, it must, after a few seconds, have attained a velocity where the piston had to break. At the maximum this must have been about 200 revolutions per minute, but only for a few revolutions, as the oiler, Cunningham, after cleaning the piston rod, had not noticed anything, and had made only a few steps when pieces of the cylinder flew up. The cover itself did not fall down until the cylinder fell, as was stated in the findings of the Liverpool Court; for this cover could only have found its way down through platform III, the greater part of which it would have taken along with it. But this platform was still intact when the first photograph was taken, only a small piece of it being missing, and this the piece which during the fall was in the way of the 24-inch steam nozzle on the L.P. cylinder, so that it was carried along.

Then the cover was not found alongside the engine, as might have been expected, if it had fallen *before* the cylinder did—somewhere near the fore-and-aft bulkhead—but way below, partly

in the crank pit among the cylinder fragments. Possibly, or probably the first piece of the piston flew upwards and lodged between the cover and the portion of the piston still secured to the rod. Then the piston would have broken immediately, for which the recess in the cylinder bottom (Fig. 6), 24 inches diamter and 2½ inches deep, would have acted as if made for the purpose, like a reservoir of destruction. If the piston-rod flange should have struck a piece here, only the size of the hand, on the next down stroke the bottom of the cylinder must have been knocked out. To do so would have required very little power, as the material was cast iron, and the power was already enormous, due to the speed of the engine. (To imagine the extraordinary circumstances attending this occurrence, we must keep in view the usual method of securing the piston with a single nut and tapered rod end. Here the nut is very small, often no larger than the lower cylindrical portion of the rod. If, with this method of securing, no part of the hub remains on the taper, then the fragments remain quietly on the bottom, and the moving piston rod can find its way through them without resistance.)

If now a piece of the cylinder bottom should get between one of the upper bearings of the connecting rod and the crosshead slipper, the cylinder would tumble down the next moment, because the connecting rod would then become a means of destruction of the worst kind.

During oscillation, the upper ends of the connecting rod describe about the crosshead pins in a horizontal direction a distance of 5½ inches, and this occurs twice in a different direction in every revolution. If anything gets between the end of the rod and the crosshead, the connecting rod then acts like a bell crank. The lower connecting-rod end turned by the full power of the H.P. and I.P. cylinders, which continue to work ahead with more than 5,000 I.H.P. (because the total power of these two cylinders working with the maximum piston speed at this moment is liberated for the purpose) and describes in the mid position (Fig. 3) the total crank path, whereas the upper end of the rod is very nearly at rest. It follows from this that an enormous pressure is put upon the crosshead, and that it must

yield to it at once; but if the crosshead is broken then this pressure is immediately transferred to the columns, and to both at the same time, and of such intensity that no material or construction can resist it. It is thus quite possible that both columns not only broke at the same moment, but that the pressure transmitted to the feet of the cylinders broke them, or at any rate started the four longitudinal fractures. Such a case cannot happen with the other arrangement of the guides, where the guide is only on one side, as a body falling down from above can find no hold, while the crosshead can generally have no side motion, so that a squeezing can not happen as in the case of the engine of the *City of Paris*.

That the connecting rod really acted in the manner described is shown by the bending of the rod to starboard (Fig. 3, in which the direction of the motion of the cranks for the forward motion is drawn); and whether the fragments fell to the right or the left of the crosshead, the result would have been the same. Of course this bending could also have been caused by the connecting rod striking something on the starboard side; but then a large impression would have been found on the concave side of the bend near the center, which was not the case; or the bearing caps might have been skewed against some obstacle. Here the result would have been seen on the bolts; but these were, as far as I could see, not much damaged. After the cylinder tumbled, the connecting rod fell towards the condenser, while the piston rod fell backwards towards the shaft. The reversing and air-pump shafts served as supports for the connecting rod. If it had been working towards the side of the bulkhead, the latter would have been entirely broken through on a level with platform I, whereas the damage to the condenser corresponds to the extreme position of the connecting rod to starboard.

The cylinder could not have fallen in the direction of its axis produced, as it was connected to the I.P. cylinder and the engine-room bulkheads, etc., in the following manner (see Figs. 3, 4 and 5): Towards the side of the fore-and-aft bulkhead by the stays *f*, receiver steam pipe *w*, the connecting bolt *b*, one of the small beams *r*, steam pipe *t*, and by two large exhaust pipes

and the reversing shaft *i*, which was connected to the cylinders by two bearings. The connections on the fore-and-aft bulkhead side could have been more easily broken than those towards the condenser, where first the two bearings and the large reversing-shaft bearing bracket on the columns of the I.P. cylinder had to break, and the falling cylinder make a turn; therefore, no doubt the flange *a* of the I.P. cylinder (towards the condenser) broke, and the two large cylinder fragments struck in such a way that the upper end of the axis fell towards the fore-and-aft bulkhead, and this agrees with the photograph.

The fragments of the columns were found somewhere near the center of the heap, between the two large pieces of the cylinder, mixed up with the fragments of the cylinder liner. The columns, therefore, fell vertically with the cylinder. The lower portions were probably broken off by the falling cylinder.

After the connecting rod continued its work of destruction for a short time a shock followed, causing the whole vessel to tremble, and everything then became quiet. The crank was found on the upper dead center, wedged tight by fragments. These finally got between the cranks and the bottom of the crank pit, and this caused the shock by which the line shafting, beginning at the L.P. cylinder, was raised up as far as the stern tube stuffing box, breaking all bearing caps on this line. That the shaft was raised $5\frac{7}{8}$ inches near the engine could be seen on the bulkheads through which the shaft goes near the thrust bearing, and at the entrance to the shaft alley. This $\frac{1}{2}$-inch plating was rolled up to a height of from $4\frac{3}{4}$ to 7 inches, and the openings thus made caused the dynamo room and the shaft alley to fill with water. How much the shaft was raised at the bearing near the stuffing box could not be ascertained. The length of the shaft thus lifted was over 98 feet, and weighed about 55 tons. To lift it eight bearing caps and their bolts had to give way, the two large caps with ten horseshoes on the thrust bearing be broken, the stuffing box split, and the $\frac{1}{2}$-inch plate of two water-tight bulkheads had to be rolled up from $4\frac{3}{4}$ to 7 inches; and besides this, there was enough power left to pierce a hole about 12 inches diameter below the crank in the $\frac{5}{8}$-inch plate of the foundation and to twist

the cranks about $\frac{1}{8}$ inch around the shaft! And all this without any appreciable swinging moment being present in proportion to the work done by the H.P. and I.P. cylinders, as only the shaft with the three cranks came into consideration. Injury to the vessel itself was only averted by the extraordinarily strong foundation. The nicks on the outer crank hubs were large, but not in proportion to the power applied to the raising of the shaft, etc.

Briefly, the occurrence would have been about as follows: 10 to 20 revolutions after the breaking of the shaft sufficed to bring about the destruction of the piston. When Cunningham saw pieces flying around, the bottom of the cylinder had probably been knocked out and the engine room filled with steam. The connecting rod acted like a bell crank during several revolutions on the falling pieces from above, pressed several pieces of the columns out, hurled them against the bulkhead and pierced it. (The piston rod could not possibly have done this.) The thin, but very hard and, superficially, very elastic material of the columns was well suited to such a process. After the columns had been thus broken in numerous pieces, the cylinder being also cracked in the direction of its axis, then a sufficiently large piece of the cylinder bottom fell on the crosshead, and the cylinder, cover and the already broken columns, fell together, and the cylinder cover, now badly damaged and with flange broken off, fell through the cylinder fragments to the bottom, while the connecting rod fell to starboard and remained there to the end. The other damage is to be explained by the play of the connecting and piston rods. The crank pit soon became filled with fragments, one of which, acting like a wedge, found its way into the clearance space between the cranks and the foundation, and thus caused the raising of the shaft, etc. Further turning of the shaft was prevented by the fragments, which filled the crank pit. With the condenser were destroyed all the large sea connections of the centrifugal pumps, and the water poured into the starboard engine room in great streams. It appears that the watertight doors of this room (one in the center bulkhead and one leading to the dynamo room) were closed a few minutes later,

after the starboard engine stopped, but the water found its way to the dynamo room and shaft alley through the opening which was made by the shaft when lifted, and to the port engine room through three large holes in the bulkhead, as well as from below, so that the port engine room was soon filled with water, while compartments Nos. 10, 11 and 12 were flooded by the suction pipes of the bilge pumps leading there, and which were not arranged for shutting off.

The evidence of this, that everything had to happen in this way, is furnished partly by the numerous large photographs which were shown to me by the Inman Co. Copies of two of them can be seen in "The Engineer" and in "Engineering," but they do not compare in size nor in accuracy with the originals.

IX.

Catastrophes seldom or never originate from a single or a simple cause, but when a number of unfavorable circumstances unite in one and the same direction and at the same moment. So it was here: faulty design, bad material and carelessness combined in a work of destruction the like of which was never seen before, and came very near sacrificing one of the most costly vessels and a thousand human lives.

It was great carelessness (1) not to take out the shaft while repairing in dock, so as to find the cause of the first abnormal wear on the bearing; (2) not to notice that the shaft, after the second wearing off, must have run unevenly and got warm long before the break occurred, and, greatest of all (3), that, as far as can be gathered from the proceedings of the Court, nobody was at the engineer's station during the racing of the engine.

It was faulty design (4) in having the shaft bearings so far apart, and (5) in having an abnormal pressure on the lignum vitæ; (6) in not having a governor on the engine; (7) in having so little compression, owing to the bad arrangement of the valves, and (8) the proportions of the L.P. piston were insufficient.

Finally, what can be charged to insufficient material is that (9) the strut bearing was rapidly worn off, possibly on account of the

imperfect condition of the propeller shaft, and (10) that the breaking of the piston occurred very rapidly.

And to this must be added the unfortunate piston-rod flange, which acted like a steam hammer on the bottom of the cylinder; and then the kind of guides and connecting rod, which, at the moment the cylinder broke, acted as a further means of destruction of the worst kind; and, to fill the measure, night came on and three large uninjured compartments filled with water, because the suction pipes of the destroyed bilge pump leading thereto were without means for shutting them off.

The accusations are so great that I feel it my duty to quote the proof of the evidence on which I base them, so that as far as possible it can be verified.

1. The proceedings of the Court, which I found most complete in the "Journal of Commerce" and the "Liverpool Daily Post" from June 16 to 25, 1890, one supplementing the other in many points.

The striking irregularities which came to light through the evidence of the engineers, and which have already been mentioned (under IV), were, as far as could be seen from the record, passed over without any action on the part of the Court. McLaren, who, instead of being on watch, was drinking tea, was heard without a word of censure.

If the sole problem of the Court was to ascertain if any person was culpable or not, those versed in law must decide. But the world expected that the cause of the accident would be cleared up by the Court, so that similar ones might be avoided in future. A true and comprehensive examination would have given valuable results on the behavior of the different materials and parts, as respects strains of such various natures as occurred during the accident, and it is to be regretted that this was neglected.

2. My own examination on board the vessel, in Laird's boiler shop, in the drydock and on the quays at Birkenhead, where the broken shafts were lying, etc. The large photographs already mentioned were of great value to me.

3. Communications from Mr. C. Busley, who arrived in Liverpool a day ahead of me, and who saw and ascertained many

things that I missed, gave me valuable hints in regard to the inspection.

The drawings, Figs. 1 to 5, are from dimensions and sketches which I made, and partly determined by the aid of the perspective views in "Engineering." Many dimensions were also given during the proceedings of the Court.

X.

What should be done to prevent such accidents has already been pointed out. It might be well, however, to define the meaning of racing. A marine engine often races without becoming disabled, but it does not follow that the speed of an engine can be increased indefinitely. In the first place it is limited by the area of the steam ports; with increasing speed, the upper line of the indicator diagram falls (from wire drawing) and the lower line rises (from increasing back pressure) until the area of the diagram becomes equal to the work of friction, upon which permanency sets in, the duration of which is governed by the capacity of the boilers to supply steam.

This permanency, that is the maximum speed of the engines, naturally takes place earlier on older engines with low piston speeds and small port areas than on those of high piston speed and large port areas; that is, all conditions being equal, an engine with small port areas can not attain such a high velocity during racing as one with large ports. It therefore follows that, in high-speed engines, the reciprocating parts, on account of the larger friction surfaces necessary, shocks, etc., must be more heavily designed than in slow-speed engines, and thus the pressure of acceleration per horse power for these parts becomes much greater. And since the clearance also is greater on account of the larger steam ports, the compression is, therefore, less than on slow-speed engines; it is clear then, that the danger in case of racing in the former must be greater than in the latter. This all has special reference to the L.P. cylinder, where the compression on marine engines is generally too small. And that is why most breaks occur to them. The H.P. and I.P. cylinders are less

subject to danger, because, on account of the high back pressure, the compression reaches a terminal pressure which almost equals that in the valve chest. Then again the weight of reciprocating parts per unit of piston area and horse power is greater than in the L.P. cylinder; and surely the tail rods on the *City of Paris* were not cut off for the reason given to me, but mainly to reduce the weight of the piston rods (about one ton for both engines) so that the motion of the engine would be a steady one, and it is also to be noted that the engine had to make from 82 to 90 revolutions, instead of the proposed 75.

The condition of a quiet motion of the L.P. engine, free from shocks, can only be obtained by compression when the cut-off varies from 0.5 to 0.75, and the initial pressure is only 1.5 atmospheres. If this can not be brought up to the necessary pressure, then the piston and connecting rods must be made hollow. By using the simple separated flat slide valves the clearance would be reduced—even for 900 feet piston speed—from 6 to 7 per cent., and this would still do for a cut-off of 0.75 with the ordinary link motion, so as to bring the compression pressure up to the valve-chest pressure. This can not be obtained with piston valves, especially if four piston valves are used, as on the L.P. cylinder of the *City of Paris;* these, then, should be discarded, and be fitted only to the H.P. cylinder.

The piston, especially of the L.P., is now generally built as a single-walled conical body of cast steel; but the distribution of the material must be such that no false stresses remain in the casting after it has cooled off.

How a piston should not be made is shown in Fig. 6. On the pistons of the United States war vessels, Laird's and others, the mass of the metal is in the conical body; hub and rim for the packing (simple, light steel rings, with self-acting springs, as on locomotive pistons) have very little material. For securing the piston to the rod the simple taper with nut is sufficient under all circumstances. Connection by means of a flange, as on the *City of Paris*, is much heavier, more costly, and is besides dangerous. All pistons that I know of on high-speed engines from first-class firms have the taper and nut.

Of the necessity for a good governor on high-speed engines, and for the utmost discipline among the engineers, to the end that any man intended for the position of engineer on fast steamers should not leave his post for a moment, no further words need be wasted.

As a loosening of the connections between piston rod, crosshead and connecting rod is not impossible even on the best of engines, provision should be made in the design of new engines so that such parts as sea valves, bilge pumps, etc., whose destruction might place the engine room under water, should not be placed in the line of flight of the connecting rod. These sea connections should also be made so that they can be closed from the main deck, and it is self-evident that all the other watertight compartments of the ship should either be connected with their own pumps, or that the bilge suction pipes leading to them should have absolutely reliable means for shutting them off (check valves), which, to judge by the record of the Court, was not the case in this instance.

PRACTICAL APPLICATION OF FORMULÆ FOR FINDING THE TURNING EFFORT ON THE CRANK SHAFT OF AN ENGINE.

By Passed Assistant Engineer B. C. Bryan, U. S. Navy.

[Read at the annual meeting.]

The object of this paper is first to reduce to a convenient form for practical application the various terms entering into the calculation of the turning effort on the crank shafts of high-speed engines. Nothing new is claimed for the methods employed in this part of the paper, and most of the work has been done, at various times, by Passed Assistant Engineer F. H. Bailey, U. S. Navy, and others. Second, to call attention to a practical way of applying these results for finding the turning moments of an engine.

I. Formulae.

We will consider the parts as all reciprocating in straight lines like the piston, crosshead and piston rod. The connecting rod will not reciprocate exactly in this way, but for nearly all practical purposes it may be taken as above, the average error being less than one per cent.

Let S = the unbalanced steam pressure on the piston = area of the piston (A) $\times$ the steam pressure per square inch at any point.

s = length of stroke in feet.

a = length of crank = $\frac{s}{2}$.

W = weight of reciprocating parts (piston, piston rod, crosshead and connecting rod) in pounds.

R = revolutions per minute.

θ and φ = angles made by the crank and the connecting rods with the axis of motion of the piston, measured as shown in Fig. 1.

b = length of connecting rod, in feet.

$\frac{b}{a} = \frac{b}{\frac{s}{2}} = n$ = length of connecting rod divided by length of crank arm.

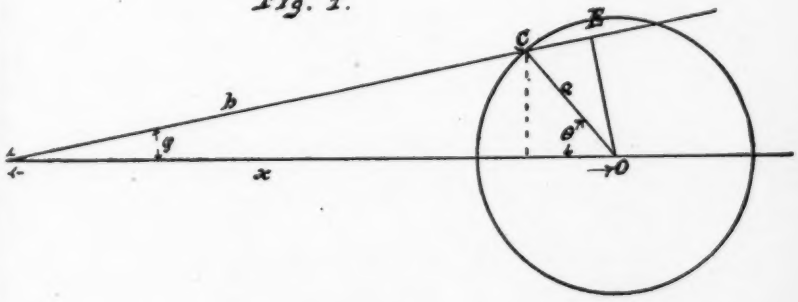
P_I = total inertia pressure, or a pressure sufficient to either retard or accelerate the piston at the required speed at any point.

$p_i = \frac{P_I}{A}$ = inertia pressure per square inch on piston.

g = gravity = 32.

x = distance from center of crosshead pin to center of engine shaft.

Fig. 1.



Now, the moment on the shaft at any point will consist of the moment produced by the steam plus that produced by the inertia,

$$= M_1 \text{ plus } M_2 = M. \qquad (1)$$

But M_1 will be equal to the total unbalanced steam pressure S resolved into its direction on the connecting rod, and multiplied by the perpendicular to its direction from the center of the shaft. Resolving this force along the rod, we have the force on the rod producing two forces—one the pressure on the slides,

and the other the force in the direction of the piston; or in other words, the force along the rod $\times \cos \varphi = S$, or force along the rod will be $S \sec \varphi$, and its arm will be the perpendicular $OE = OC \sin OCE = \frac{s}{2} \sin (\theta + \varphi)$.

$$\therefore \quad M_1 = Ss \left[\frac{\sin (\theta + \varphi) \sec \varphi}{2} \right]. \tag{2}$$

In considering the inertia, x representing the distance from the center of the crosshead pin to the center of the engine shaft at any time, dx will represent the velocity of the piston at any point x, d^2x the acceleration at the same point, and $d\theta$ the angular velocity of the crank, which is assumed as constant.

From the above figure, we readily see that

$$x = a \cos \theta + b \cos \varphi, \tag{3}$$

also $\qquad a \sin \theta = b \sin \varphi$, or $\sin \varphi = \frac{a}{b} \sin \theta$.

Squaring both sides, and subtracting from (1)

$$1 - \sin^2 \varphi = 1 - \frac{a^2}{b^2} \sin^2 \varphi.$$

Clearing of fractions and extracting the square root,

$$b \cos \varphi = \sqrt{b^2 - a^2 \sin^2 \theta}.$$

Substituting in (3), we have

$$x = a \cos \theta + \sqrt{b^2 - a^2 \sin^2 \theta}.$$

Differentiating,

$$dx = -a \sin \theta \, d\theta - \frac{a^2 \sin \theta \cos \theta \, d\theta}{\sqrt{b^2 - a^2 \sin^2 \theta}}$$

$$= -a \left(\sin \theta + \frac{a \sin \theta \cos \theta}{\sqrt{b^2 - a^2 \sin^2 \theta}} \right) d\theta.$$

Differentiating again, and remembering that $d\theta$ is constant,

$$d^2x = -a \left(\cos \theta + \frac{a \cos^2 \theta - \sin^2 \theta}{\sqrt{b^2 - a^2 \sin^2 \theta}} + \frac{a^3 \sin^2 \theta \cos^2 \theta}{\sqrt{(b^2 - a^2 \sin^2 \theta)^3}} \right) d\theta^2 \tag{4}$$

Now force equals acceleration multiplied by weight and divided by gravity, or $P_1 = d^2x \frac{W}{g}$, and $p_i = \frac{P_1}{A} = d^2x \frac{W}{Ag}$.

$d\theta$ = velocity of crank pin in feet per second divided by the radius of crank circle.

The crank pin travels $2\pi a = \pi s$ for each revolution.

The piston travels $4a$ for each revolution.

R = revolutions per minute, $\frac{R}{60}$ = revolutions per second.

$\frac{\pi s \times R}{60}$ = velocity of crank pin in feet per second.

$$d\theta = \frac{\pi s \times R}{60} \times \frac{2}{s} = \frac{\pi R}{30}, \text{ and } d\theta^2 = \frac{\pi^2 R^2}{900}.$$

Multiplying both members of equation (4) by $\frac{W}{Ag}$, and substituting for $d\theta^2$ its value, and for $\frac{b}{a}$ its value n, or $b = na$ and $a = \frac{s}{2}$ we have

$$p_i = d^2x \times \frac{W}{Ag} = -\left\{\cos\theta + \frac{\cos^2\theta - \sin^2\theta}{\sqrt{n^2 - \sin^2\theta}} + \frac{\sin^2\theta \cos^2\theta}{\sqrt{(n^2 - \sin^2\theta)^3}}\right\} \quad (5)$$

Now $M_2 = p_i \times A \times S \sec\varphi \times$ arm OE, as before;

$$M_2 = p_i \times A \times Ss\left[\frac{\sin(\theta+\varphi)(\sec\varphi)}{2}\right]; \text{ substituting in (5)}$$

$$M_2 = -\left\{\cos\theta + \frac{\cos^2\theta - \sin^2\theta}{(n^2 - \sin^2\theta)^{\frac{1}{2}}} + \frac{\sin^2\theta\cos^2\theta}{(n^2 - \sin^2\theta)^{\frac{3}{2}}}\right\}$$

$$\sin(\theta+\varphi)\sec\varphi \times \frac{WR^2s^2\pi^2}{115{,}200}. \quad (6)$$

The total moment of the shaft will be (adding (2) and (6)) $M = M_1 + M_2$

$$= Ss\left[\frac{\sin(\theta+\varphi)\sec\varphi}{2}\right] + WR^2s^2\left[-\sin(\theta+\varphi)\sec\varphi\right.$$

$$\left.\times \frac{\pi^2}{115{,}200} \times \left\{\cos\theta + \frac{\cos^2\theta - \sin^2\theta}{(n^2 - \sin^2\theta)^{\frac{1}{2}}} + \frac{\sin^2\theta\cos^2\theta}{(n^2 - \sin^2\theta)^{\frac{3}{2}}}\right\}\right]. \quad (7)$$

This may be written $M = Ss \times P + WR^2s^2 \times \omega$, where P and ω represent the expressions in brackets.

Then, $M =$ unbalanced pressure $\times$ stroke of piston $\times P +$ weight of parts $\times K^2 \times$ stroke$^2 \times \omega$.

It is evident that all engines having the same ratio of crank to connecting rod will have the same value of n. This being fixed, the only quantities that vary for different engines will be weight W, unbalanced pressure, S, and stroke, s. The values of n used in naval engines vary but little, and table No. I has been made for different values of n, showing the values of P and ω corresponding to each value of θ, the crank angle, for each 15°. Now, multiplying the value Ss and WR^2s^2 by the corresponding multiplier in the table, the values of the moment due to steam pressure, M_1, and that due to inertia, M_2, may be found, and their sum is the total moment on the shaft. Care must be taken to use the correct sign for the inertia effort as given in the table.

II. Practical Application of the Formulæ.

To apply this table to finding the curve of turning moments of an engine in designing the parts, we may proceed in two ways. First, if the engine is one of novel design and differs radically from any that we have cards from, it would be necessary to lay down the theoretical card and take ordinates from it for our unbalanced pressures; but as this would be a very unusual case in the present advanced state of mechanical engineering, and as changes from low to higher pressures are made gradually, we may (2d,) take the card of some similar engine that is known to give good results, but using a different pressure, and lay down from it a curve of expected unbalanced pressures. A sufficient approximation can in this way be obtained in almost any case.

To illustrate this method, I have selected four sets of cards taken from the port engine of the U. S. S. *Detroit*. It was found a difficult matter to select a set of cards that I could feel sure were all right, as some error was usually noticed that made them unfit for comparison, either from steam being admitted to the receivers or from the links of one or more cylinders not be-

ing in full gear, the work not being equally divided between the cylinders, etc. Even after selecting my cards I find that the I.P. cylinder is doing less work than the others, and that the H.P. links were run in a little from full gear.

However, these cards will do for the sake of illustration, and I have reduced them to apply to another engine using a higher pressure, as follows:

	Detroit.	New engine.
Diameter of H.P. cylinders, inches, . .	26½	18½
I.P. cylinders, inches, . .	39	29½
L.P. cylinders, inches, . .	63	2 of 33¼
Stroke, inches, ...	26	20
Length of connecting rod, inches, . .	52	45
Designed boiler pressure, ...	160	250
Designed revolutions, ...	185	330

Each card was divided up into the proper number of ordinates corresponding with the table for multipliers, by laying off every 15 degrees on the crank circle, and taking the length of the connecting rod to same scale as a radius, and striking an arc to cut the center line of the cylinder. These ordinates were measured in the usual manner to the scale of the indicator spring from the line of zero pressure, and the reading of the top and bottom lines, or absolute pressures for each ordinate, are placed separately in lines 1, 2, 6, 7, 11 and 13 in the blank I have prepared (Table II). It is well to note that (the unbalanced pressure being what we are after) the reading to the top line of one card is placed in line 1, and the reading to the bottom line of the card for the other end of the cylinder on the same ordinate is placed in line 2, as the difference of these readings gives the unbalanced pressures at the position of piston represented by any ordinate.

Now, it must be remembered that the engine to which we wish to apply these cards is supposed to be similar to the one that I have taken cards from, that the clearance is about the same per cent., cut off the same, and that what we wish to ascertain is what unbalanced pressures we may expect to get from an engine of the same type, but of different cylinder ratio, and us-

ing steam of a different pressure. We will now consider the cylinders in regular order.

1st. *High-pressure cylinder.*—These engines being exactly similar, the expansion line should be in the ratio of the absolute pressures used. The boiler pressure at the time the card given in Table II was taken was 162 lbs. per gauge = 176.7 lbs. absolute; in the new engine it is to be 250 lbs. per gauge = 264.7 absolute. The first step is then to correct our expansion line or absolute reading of the top line of the card found in line 1, by multiplying each ordinate by the ratio of the absolute pressures, viz: $\frac{264.7}{176.7}$, the results being placed in line 3.

The steam on the bottom line has expanded an amount due to the ratio between the space it occupied in the H.P. cylinder before expansion and the space it occupies in the I.P. cylinder; the engines being similar, this is represented in each case by the ratio of the H.P. to the I.P. cylinders (the point of cut-off being the same in both engines), and our multiplier for correcting this bottom line will be the ratio of the absolute pressure multiplied by the ratio of the cylinder ratios, or, if ratio of $\frac{\text{I.P. cyl.}}{\text{H.P. cyl.}}$ of *Detroit* $= r$, and those of the new engine $= r'$, we have a multiplier for the bottom line $\frac{264.7}{176.7} \times \frac{r}{r'}$, or in this case $= \frac{264.7}{176.7} \times \frac{2.165}{2.556}$ $= 1.275$.

The result of multiplying each ordinate in line 2 by this multiplier is found in line 4, and these subtracted from the results in line 3, give the expected unbalanced pressures, line 5. From these results the unbalanced pressure curves $H, H_1, \ldots H$ in Plate I, are laid down for the high-pressure cylinder, the abscissæ representing divisions of the crank circles and the ordinate at any point representing the unbalanced steam pressure at that point.

2d. *I.P. Cylinder.*—The top reading of the card for this cylinder, line 6, will have the same multiplier as the bottom line of the H.P. cylinder, viz: $\frac{P'}{P} \times \frac{r}{r'}$; the corrected ordinates for this

line are tabulated in line 8. Reasoning in the same way, the bottom line of this cylinder will have as a multiplier the ratios of $\frac{\text{L.P. cyl.}}{\text{H.P. cyl.}} = R$, instead of ratio of $\frac{\text{I.P. cyl.}}{\text{H.P. cyl.}}$. Hence multiplier will be $\frac{P'}{P} \times \frac{R}{R'}$, and corrections for this line will be found tabulated in line 9. The difference between lines 8 and 9 will, as before, give the expected unbalanced pressure for each ordinate of the I.P. cylinder, and these laid off as for the H.P. cylinder give a similar curve for the I.P. cylinder. (Plate I. $I, I, \ldots I$.)

3d. *L.P. Cylinder.*—The readings of the top line of this cylinder, line 11, will have the same multiplier for correction as the bottom line of the I.P. cylinder, and these readings corrected are tabulated in line 12. The bottom line of this cylinder will have no correction, but will depend upon the vacuum carried; that in the *Detroit* being 25½ inches per gauge, we may take that as being as good as we ought to expect; so the readings of the card for this line are tabulated in line 13, and these results subtracted from the results in line 12 give the unbalanced pressure that may be expected (line 14), from which the curve for this cylinder may be constructed as before. (Plate I. $L, L, \ldots L$.)

Lines 15, 16 and 17, Table II, give the average results of four sets of cards taken from the *Detroit*, and reduced in this way to apply to the new engine.

N. B.—The cards in Plate I are constructed from these average results.

The part on the left hand and above the axis is taken to represent pressures on the top of the piston, or for the down stroke, and the part below the axis and to the right represents pressures on the bottom of the piston. The axis represents zero pressure, and all pressures below it for the *top* of the cylinder are minus pressures; also, all above it for the *bottom* of the cylinder are minus.

It is evident that these curves should be fair curves without points of deflection, and where such occur mistakes may be looked for. I find these mistakes caused primarily by vibration of the indicator, which, though very slight in the original, are

magnified in correcting to a higher pressure. These points, though marked for single cards, are pretty well eliminated by taking the average of a number of cards.

Having now our curves of unbalanced pressures, we may proceed to construct our curves of twisting moments. But first we must consider the effect of gravity on the weight of the moving parts. This effect will be constant and equal to a steam pressure per square inch on each piston equivalent to the weight of the parts divided by the area of the piston. The weight of the moving parts in our new engine will be as follows:

H.P. = 1,337 pounds; I.P. = 1,430 pounds; L.P. = 900 pounds.

Hence the pressure per square inch produced by this weight will be for H.P. $= \frac{1,337}{268.8} = 4.97$; I.P. $= \frac{1,430}{683.494} = 2.19$, and L.P. $= \frac{900}{868.3} = 1.04$ pounds. This pressure is to be added to the pressures on top of the piston, and subtracted from the pressures on the bottom, which may be done by drawing a line, $A'B'$, parallel to the axis AB, Plate I, and at a distance representing such pressure from it, and measuring our ordinates from this line instead of from the original axis AB. (In Plate I, $A'B'$ is drawn for the H.P. cylinder only.)

These ordinates are measured and results placed in columns 1, 2 and 3 in Table III.

In columns 4, 5 and 6 are placed the results of multiplying the numbers in columns 1, 2 and 3, respectively, by the area of each cylinder and by its stroke in feet.

Column 4 = column 1 × 448; (area $18\frac{1}{2} \times 20$ inches = 268.8 × $1\frac{2}{3}$ feet = 448).

Column 5 = column 2 × 1,139.157; (area $29\frac{1}{2} \times 20$ inches = 683.494 × $1\frac{2}{3}$ feet = 1,139.157).

Column 6 = column 3 × 1,447.2; (area $33\frac{1}{4} \times 20$ inches = 868.3 × $1\frac{2}{3}$ inches = 1,447.2).

Column 7 contains the multipliers for moments due to pressures taken from Table I for a value of $n = 4.5$ (crank = 10 inches and connecting rod = 45 inches).

Columns 8, 9 and 10 contain the results of multiplying columns 4, 5 and 6 by the multipliers in column 7.

In columns 11, 12, 13 and 14 these moments are arranged according to regular order of cylinders: the I.P. cylinder is at 180° to the H.P., and the first L.P. is 90° behind the H.P., and the second L.P. at 180° from the first L.P.

In column 15 we have the sums of the four cylinders, or the resultant moment due to the pressures alone for each 15° of the crank circle.

We must now find the effect of the inertia of the reciprocating parts. I have taken the full weight of the connecting rod, crosshead, piston rod and piston and rings for this as given above.

The values of WR^2s^2 will be—

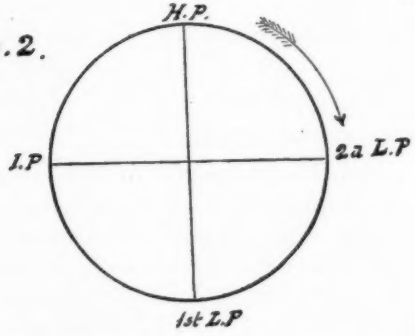
$$\text{For H.P.} = 1{,}337 \times (330)^2 \times (1.667)^2 = 404{,}442{,}500.$$
$$\text{I.P.} = 1{,}430 \times (330)^2 \times (1.667)^2 = 432{,}575{,}000.$$
$$\text{L.P.} = 900 \times (330)^2 \times (1.667)^2 = 272{,}250{,}000.$$

In column 16 we have the tabulated multipliers taken from Table I for value of $n = 4.5$, and in columns 17, 18 and 19 we have the moments due to inertia found by multiplying the value of WR^2s^2 for each cylinder by its proper multiplier for each ordinate. It is well to observe that the inertia does not affect the total or mean value of the moment, as the amount of work lost in acceleration during the first part of the stroke is given up during the latter part. It may, however, affect very seriously the maximum moment. In columns 20, 21, 22, 23, the values of the combined moments for pressure and inertia are tabulated in regular order of cylinders, and column 24 gives the resultant moment for both pressures and inertia.

Plate II gives the curves of moments plotted on the development of the H.P. crank circle; the dotted lines show the curves for pressure alone without considering inertia, and the full lines show the total moments for both pressures and inertia. The curves *R*, *R*, *R*,....*R*, are the resultant curves, the dotted one for pressures alone as before, and the full line for pressures and inertia combined.

The relative positions of the cranks may now be shifted about until certain positions are found that will give a minimum value of the maximum resultant moment, and other things being equal these will be the best angles at which to place the cranks. But in doing this care must be taken that the lateral vibrations of the engine are not increased, and that a bad angle of cranks for reversing and starting the engines is not taken. The curve *R, R, R,* shown in broken line, which is the result with the cranks placed in the following order: I.P. 90° behind H.P., and 1st L.P. 90° behind I.P., and 2d L.P. 90° behind 1st L.P., or as shown in the diagram.

Fig. 2.



This arrangement gives an excellent moment, but it is believed that the lateral vibration would be greater at a high speed.

The mean moment of the original arrangement of cranks is 42,213 foot pounds, which under the above conditions would give us as the horse power of our engine $\frac{42{,}213 \times 2\pi \times 330}{33{,}000} =$ 2,650, which is about 10 per cent. more than is expected of the engine in question. The ratio of maximum to mean moment is $\frac{55{,}450}{42{,}213} = 1.313.$

With the second arrangement of cranks this ratio becomes: $\frac{48{,}390}{42{,}213} = 1.146.$

TABLE I.

$n = \frac{\text{Length of Connecting Rod}}{\text{Length of Crank}}$. P = Multiplier for Pressure. ω = Multiplier for Inertia.

Angle of crank from top.	$n = 3.5$		$n = 4.0$		$n = 4.5$		$n = 5.0$		$n = \infty$	
θ	ω	P.	ω	P.	ω	P.	ω	P.	ω	P.
15° 345°	$\mp .00003441$	.1652	$\mp .00003257$	.1606	$\mp .00003125$	.1572	$\mp .00003019$	.1545	$\mp .00001973$	.1294
30° 330°	$\mp .00005426$	.3125	$\mp .00005193$	.3044	$\mp .00005012$	.2984	$\mp .00004867$	.2934	$\mp .00003710$	.2500
45° 315°	$\mp .00005214$	.4264	$\mp .00005084$	.4170	$\mp .00004985$	.4096	$\mp .00004911$	.4040	$\mp .00004235$	.3536
60° 300°	$\mp .00003044$	.4969	$\mp .00003141$	.4885	$\mp .00003214$	.4820	$\mp .00003270$	.4769	$\mp .00003710$	.4330
75° 285°	$\mp .00000264$	.5200	$\mp .000003256$	.5150	$\mp .00000548$	.5115	$\mp .00000722$	.5085	$\mp .00001975$	.4830
90° 270°	$\pm .00002555$	.5000	$\pm .0c002213$	.5000	$\pm .00001953$	.5000	$\pm .00001749$	.5000	$\pm .00000000$	.5000
105° 255°	$\pm .00003934$	.4459	$\pm .00003714$	.4507	$\pm .00003545$	.4544	$\pm .00003409$	.4575	$\pm .00001975$	.4830
120° 240°	$\pm .00004067$	.3692	$\pm .00004044$	.3775	$\pm .00004022$	.3840	$\pm .00004000$	.3889	$\pm .00003710$	.4330
135° 225°	$\pm .00003372$	.2806	$\pm .00003494$	.2900	$\pm .00003589$	.2973	$\pm .00003661$	.3030	$\pm .00004235$	.3536
150° 210°	$\pm .00002305$	.1875	$\pm .00002467$	.1953	$\pm .00002598$	.2015	$\pm .00002691$	.2065	$\pm .00003710$	.2500
165° 195°	$\pm .00001150$	.0937	$\pm .00001259$	.0982	$\pm .00001345$	.1015	$\pm .00001417$	.1048	$\pm .00001973$	.1294

PRESSURE TAKEN FROM INDICATOR CARDS OF U. S. S. *DETRO*

U. S. S *Detroit:* Cylinders $26\frac{1}{2} \times 39 \times 63 \times 26$ inches stroke. Steam pressure in boiler = 162 lbs.

New Engine: Cylinders $18\frac{1}{2} \times 29\frac{1}{2} \times 2$ of $33\frac{1}{4} \times 20$ inches stroke. Steam = 250 lbs.

Degrees of Crank Circle from Top	0°	15°	30°	45°	60°	75°	90°
1. Absolute Pressure, Card Reading, Top Line = P	150.7	148.7	149.7	142.7	137.7	134.7	130.9
2. Absolute Pressure, Card Reading, Bottom Line = p	74.7	64.7	61.7	61.7	59.7	58.7	59.7
3. Reduced Pressure, Top line = $P \times 264.7 \div 176.7 = 1.498P$	225.7	222.7	224.2	213.8	206.3	201.8	196.1
4. Reduced Pressure, Bottom Line = $p \times 264.7 \div 176.7 \times 2.165 \div 2.556 = 1.275p$	95.6	82.5	78.7	78.7	76.1	74.9	76.1
5. Difference, or Unbalanced Pressure	130.1	140.2	145.5	135.7	130.2	126.9	120.0
6. Absolute Pressure, Card Reading, Top Line = P	54.7	60.7	57.7	57.1	56.7	56.7	53.7
7. Absolute Pressure, Card Reading, Bottom Line = p	29.7	24.7	23.7	23.2	23.2	23.2	23.7
8. Reduced Pressure, Top Line = $P \times 264.7 \div 176.7 \times 2.165 \div 2.556 = 1.275P$	69.7	77.4	73.6	72.8	72.3	72.3	68.5
9. Reduced Pressure, Bottom Line = $p \times 264.7 \div 176.7 \times 5.651 \div 6.461 = 1.31p$	38.9	32.4	31.0	30.3	30.3	30.3	31.0
10. Difference, or Unbalanced Pressure	30.8	45.0	42.6	42.5	42.0	42.0	37.5
11. Absolute Pressure, Card Reading, Top Line, = P	9.70	13.70	20.95	20.70	19.20	18.95	18.7
12. Reduced Pressure, Top Line = $P \times 264.7 \div 176.7 \times 5.651 \div 6.461 = 1.31P$	12.70	17.95	27.44	27 12	25.15	24.82	24.5
13. Absolute Pressure, Card Reading, Bottom Line	9.70	6.70	5.95	5.20	4.95	4.70	4.7
14. Difference, or Unbalanced Pressure	3.00	11.25	21.49	21.92	20.20	20.12	19.8
					Average of Four		
15. High-Pressure Cylinder	126.2	138.3	140.4	134.1	128.0	121.5	117.6
16. Intermediate-Pressure Cylinder	33.7	43.5	42.1	41.2	40.5	40.0	37.7
17. Low-Pressure Cylinder	7.14	13.49	20.42	20.79	19.26	19.02	18.9

TABLE II.

TROIT.—REDUCED TO APPLY TO ENGINE USING 250 POUNDS BOILER PRESSURE.

s.

Ratios of Cylinders: { *Detroit*..... $\frac{\text{I.P.}}{\text{H.P.}} = 2.165$ $\frac{\text{L.P.}}{\text{H.P.}} = 5.651$; New Engine..... $\frac{\text{I.P.}}{\text{H.P.}} = 2.543$ $\frac{\text{L.P.}}{\text{H.P.}} = 6.461$

H. P. Cylinder.

90°	105°	120°	135°	150°	165°	180°	180°	195°	210°	225°	240°	255°	270°	285°	300°	315°	330°
130.9	128.7	124.7	117.7	98.7	85.7	69.7	149.7	150.7	148.7	147.7	146.7	143.7	134.7	130.7	117.7	106.7	100.7
59.7	63.7	65.7	68.7	71.7	94.7	135.7	69.7	56.7	53.7	53.7	53.7	57.7	60.7	64.7	65.7	74.7	90.7
196.1	192.8	186.8	176.3	147.8	128.4	104.4	224.2	225.7	222.7	221.3	219.7	215.3	201.7	195.8	176.3	159.8	150.8
76.1	81.2	83.8	87.6	91.5	120.8	173.0	88.9	72.3	68.5	68.5	68.5	73.6	77.4	82.5	83.8	95.3	115.7
120.0	111.6	103.0	88.7	56.2	7.6	—68.6	135.3	153.4	154.2	152.8	151.2	141.7	124.3	113.3	92.5	64.5	35.1

I. P. Cylinder.

53.7	50.7	47.7	42.7	39.2	34.7	27.7	62.7	61.7	60.7	60.7	59.7	55.2	52.2	49.2	44.2	40.2	38.2
23.7	25.2	26.7	26.7	28.2	38.7	49.7	27.7	23.7	22.7	22.7	22.7	24.2	24.7	26.7	26.7	27.7	35.2
68.5	64.6	60.8	54.4	50.0	44.2	35.3	79.9	78.7	77.4	77.4	76.1	70.4	66.6	62.7	56.4	51.3	48.7
31.0	33.0	35.0	35.0	36.9	37.6	65.1	36.3	31.0	29.7	29.7	29.7	31.7	32.4	35.0	35.0	36.3	46.1
37.5	31.6	25.8	19.4	13.1	6.6	—29.8	43.6	47.7	47.7	47.7	46.4	38.7	34.2	27.7	21.4	15.0	2.6

L. P. Cylinder.

18.70	17.95	16.45	13.70	12 45	10.70	9.70	18 20	22.20	21.45	20.70	20.70	20.70	19.70	18.20	15.20	13.45	12.70
24.50	23.51	21.51	17.95	16.31	14.02	12.70	23.84	29 08	28.10	27.12	27.12	27.12	25.81	23.84	19.91	17.62	16.64
4.70	4.70	4.45	4.45	6.20	10.70	18.20	8.20	7.20	6.20	5.70	4.95	4.45	4.20	4.20	4.20	5.95	7.70
19.80	18.81	17.06	13.50	10.11	— 3.32	— 5.50	15.64	21.88	21.90	21.42	22.17	22.67	21.61	19.64	15.71	11.67	8.94

our Sets of Cards Treated as Above.

117.6	108.5	99.6	83.3	56.8	55.0	—57.9	139.8	148.8	147.7	145.5	144.0	135.6	124.0	110.4	90 1	65.1	33.5
37.7	30.8	25.7	18.2	11.1	—10.6	—33.0	42.6	48.4	48.4	47.5	46.4	39.3	33.4	27.8	20.9	13.3	2.2
18.94	18.21	16.11	11.36	9.90	45.4	— 4.56	15.29	20.90	21.38	21.16	20.91	20.18	18.78	17.32	16.00	11.20	8.20

TABLE II

PRESSURE TAKEN FROM INDICATOR CARDS OF U. S. S. *DETROIT*.—REDUCE

U. S. S *Detroit:* Cylinders $26\frac{1}{2} \times 39 \times 63 \times 26$ inches stroke. Steam pressure in boiler = 162 lbs.

New Engine: Cylinders $18\frac{1}{2} \times 29\frac{1}{2} \times 2$ of $33\frac{1}{4} \times 20$ inches stroke. Steam = 250 lbs.

H. P. Cylinder

Circle from Top	0°	15°	30°	45°	60°	75°	90°	105°	120°	
sure, Card Reading, Top Line = P	150.7	148.7	149.7	142.7	137.7	134.7	130.9	128.7	124.7	1
sure, Card Reading, Bottom Line = p	74.7	64.7	61.7	61.7	59.7	58.7	59.7	63.7	65.7	
sure, Top line = $P \times 264.7 \div 176.7 = 1.498P$	225.7	222.7	224.2	213.8	206.3	201.8	196.1	192.8	186.8	1
sure, Bottom Line = $p \times 264.7 \div 176.7 \times 2.165 \div 2.556 = 1.275p$	95.6	82.5	78.7	78.7	76.1	74.9	76.1	81.2	83.8	
Unbalanced Pressure	130.1	140.2	145.5	135.7	130.2	126.9	120.0	111.6	103.0	
									I. P. Cylinder.	
ure, Card Reading, Top Line = P	54.7	60.7	57.7	57.1	56.7	56.7	53.7	50.7	47.7	
ure, Card Reading, Bottom Line = p	29.7	24.7	23.7	23.2	23.2	23.2	23.7	25.2	26.7	
sure, Top Line = $P \times 264.7 \div 176.7 \times 2.165 \div 2.556 = 1.275P$	69.7	77.4	73.6	72.8	72.3	72.3	68.5	64.6	60.8	
ure, Bottom Line = $p \times 264.7 \div 176.7 \times 5.651 \div 6.461 = 1.31p$	38.9	32.4	31.0	30.3	30.3	30.3	31.0	33.0	35.0	
Unbalanced Pressure	30.8	45.0	42.6	42.5	42.0	42.0	37.5	31.6	25.8	
									L. P. Cylinder.	
re, Card Reading, Top Line, = P	9.70	13.70	20.95	20.70	19.20	18.95	18.70	17.95	16.45	
re, Top Line = $P \times 264.7 \div 176.7 \times 5.651 \div 6.461 = 1.31P$	12.70	17.95	27.44	27 12	25.15	24.82	24.50	23.51	21.51	
re, Card Reading, Bottom Line	9.70	6.70	5.95	5.20	4.95	4.70	4.70	4.70	4.45	
Unbalanced Pressure	3.00	11.25	21.49	21.92	20.20	20.12	19.80	18.81	17.06	
									Average of Four Sets of Cards T	
Cylinder	126.2	138.3	140.4	134.1	128.0	121.5	117.6	108.5	99.6	8
essure Cylinder	33.7	43.5	42.1	41.2	40.5	40.0	37.7	30.8	25.7	
ylinder	7.14	13.49	20.42	20.79	19.26	19.02	18.94	18.21	16.11	

II.

UCED TO APPLY TO ENGINE USING 250 POUNDS BOILER PRESSURE.

Ratios of Cylinders: *Detroit*..... $\frac{\text{I.P.}}{\text{H.P.}} = 2.165$ $\frac{\text{L.P.}}{\text{H.P.}} = 5.651$

New Engine..... $\frac{\text{I.P.}}{\text{H.P.}} = 2.543$ $\frac{\text{L.P.}}{\text{H.P.}} = 6\,461$

nder.

°	135°	150°	165°	180°	180°	195°	210°	225°	240°	255°	270°	285°	300°	315°	330°	345°	360°
7	117.7	98.7	85.7	69.7	149.7	150.7	148.7	147.7	146.7	143.7	134.7	130.7	117.7	106.7	100.7	92 7	74.7
7	68.7	71.7	94.7	135.7	69.7	56.7	53.7	53.7	53.7	57.7	60.7	64.7	65.7	74.7	90.7	110.7	124.7
8	176.3	147.8	128.4	104.4	224.2	225.7	222.7	221.3	219.7	215.3	201.7	195.8	176.3	159.8	150.8	138.8	111.9
8	87.6	91.5	120.8	173.0	88.9	72.3	68.5	68.5	68.5	73.6	77.4	82.5	83.8	95.3	115.7	141.2	159.0
°	88.7	56.2	7.6	—68.6	135.3	153.4	154.2	152.8	151.2	141.7	124.3	113.3	92.5	64.5	35.1	— 2.4	—47.1
der.																	
7	42.7	39.2	34.7	27.7	62.7	61.7	60.7	60.7	59.7	55.2	52.2	49.2	44.2	40.2	38.2	35.7	29.7
7	26.7	28.2	38.7	49.7	27.7	23.7	22.7	22.7	22.7	24.2	24.7	26.7	26.7	27.7	35.2	43.7	54.7
8	54.4	50.0	44.2	35.3	79.9	78.7	77.4	77.4	76.1	70.4	66.6	62.7	56.4	51.3	48.7	45.5	37.9
°	35.0	36.9	37.6	65.1	36.3	31.0	29.7	29.7	29.7	31.7	32.4	35.0	35.0	36.3	46.1	57.2	71.7
8	19.4	13.1	6.6	—29.8	43.6	47.7	47.7	47.7	46.4	38.7	34.2	27.7	21.4	15.0	2.6	—11.7	—33.8
der.																	
35	13.70	12 45	10.70	9.70	18 20	22.20	21.45	20.70	20.70	20.70	19.70	18.20	15.20	13.45	12.70	11.70	9.70
31	17.95	16.31	14.02	12.70	23.84	29 08	28.10	27.12	27.12	27.12	25.81	23.84	19.91	17.62	16.64	15.33	12.70
35	4.45	6.20	10.70	18.20	8.20	7.20	6.20	5.70	4.95	4.45	4.20	4.20	4.20	5.95	7.70	8.70	9.20
96	13.50	10.11	— 3.32	— 5.50	15.64	21.88	21.90	21.42	22.17	22.67	21.61	19.64	15.71	11.67	8.94	6.63	3.50
ds Trented as Above.																	
5	83.3	56.8	55.0	—57.9	139.8	148.8	147.7	145.5	144.0	135.6	124.0	110.4	90 1	65.1	33.5	— 3.42	—41.8
7	18.2	11.1	—10.6	—33.0	42.6	48.4	48.4	47.5	46.4	39.3	33.4	27.8	20.9	13.3	2.2	—14.3	—29.1
1	11.36	9.90	45.4	— 4.56	15.29	20.90	21.38	21.16	20.91	20.18	18.78	17.32	16.00	11.20	8.20	5.54	2.51

TABLE

Moments due to steam pressures.

Degrees of crank circle from top.	Unbalanced pressure. A			Unbalanced pressure × area of piston × stroke in feet. B.			Multiplier for moment due to steam. P.	Moment due to steam. C = B × P.			Moment due to steam (arr order of cylind		
	1.	2.	3.	4.	5.	6.	7.	8.	9.	10.	11.	12.	1
	H.P.	I.P.	L.P.	H.P.	I.P.	L.P.		H.P.	I.P.	L.P.	H.P.	I.P.	1st
Top	.....	.....	.....	.....	.....	.....	0	0	0	0	0	0	128
15°	143.1	45.7	14.99	64065.	52060.	21690.	.1572	10070.	8184.	3410.	10070.	5342.	120
30°	145 4	44 3	21 42	65120.	50465.	31000.	.2984	19430.	15060	9250.	19430.	10600.	104
45°	139.1	43.4	21 79	62320.	49440.	31530.	.4096	25530.	20250.	12910.	25530.	15340	60
60°	133 1	42.7	20.26	59630.	48640.	29320.	.4820	28740.	23450.	14130	28740.	19330.	31
75°	126.5	42.2	20.02	56670.	48070	28970.	.5115	28990.	24590.	14820.	28990.	19200.	12
90°	122.6	39.9	19.94	54920.	45450.	28860.	.5000	27460.	22725.	14430.	27460.	17770.	0
105°	113 5	33.0	19.21	50850.	37590.	27800.	.4544	23100.	17080.	12630.	23100.	14920.	34
120°	104.6	27.9	17.11	46860.	31780.	24760.	.3840	17990.	12200.	9507.	17990.	10270.	92
135°	88.3	20.4	12.36	39560.	23240	17890.	.2973	11760.	6910	5319.	11760.	5181.	129
150°	61.8	13.3	10.90	27690.	15150.	15780.	.2015	5580.	3053.	3180.	5580.	0	141
165°	10.5	— 8 4	5.45	4704.	— 9568.	7887.	.1015	477.	— 971.	800.	477.	— 2166.	148
180°	.....	.....	.....	.....	.....	.....	0	0	0	0	0	0	144
195°	143.8	46.2	19.90	64430.	52630.	28800.	.1015	6540.	5342.	2923.	6540.	8184.	126
210°	142.7	46.2	20.38	63930.	52630.	29490.	.2015	12880.	10600.	5942.	12880.	15060.	95
225°	140.5	45.3	20.16	62940.	51600.	29170	.2973	18710.	15340.	8672.	18710.	20250.	53
240°	139.0	44.2	19.91	62260.	50350.	28810.	.3840	23910.	19330.	11060.	23910.	23450.	31
255°	130.6	37.1	19.18	58510.	42260.	27760.	.4544	26590.	19200.	12610.	26590.	24590.	8
270°	119.0	31.2	17.78	50930.	35540.	25730.	.5000	25465	17770.	12865.	25465.	22725.	0
285°	105.4	25.6	16.32	47220.	29160.	23620.	.5115	24150.	14920.	12080.	24150.	17080.	29
300°	85.1	18.7	15.00	38125.	21300.	21710.	.4820	18380.	10270.	10460.	18380.	12200.	59
315°	60.1	11.1	10 20	26930.	12650.	14760.	.4096	11030.	5181.	6045.	11030.	6910.	86
330°	28.5	0	7.20	12770.	0	10420.	.2984	3810.	0	3109.	3810.	3053.	110
345°	— 8.4	—12.1	4.54	— 3763.	—13780.	8018.	.1572	— 592.	—2166.	1260.	— 592.	— 971.	126
360°	.....	.....	.....	.....	.....	.....	.....	0	0	0	0	0	.....

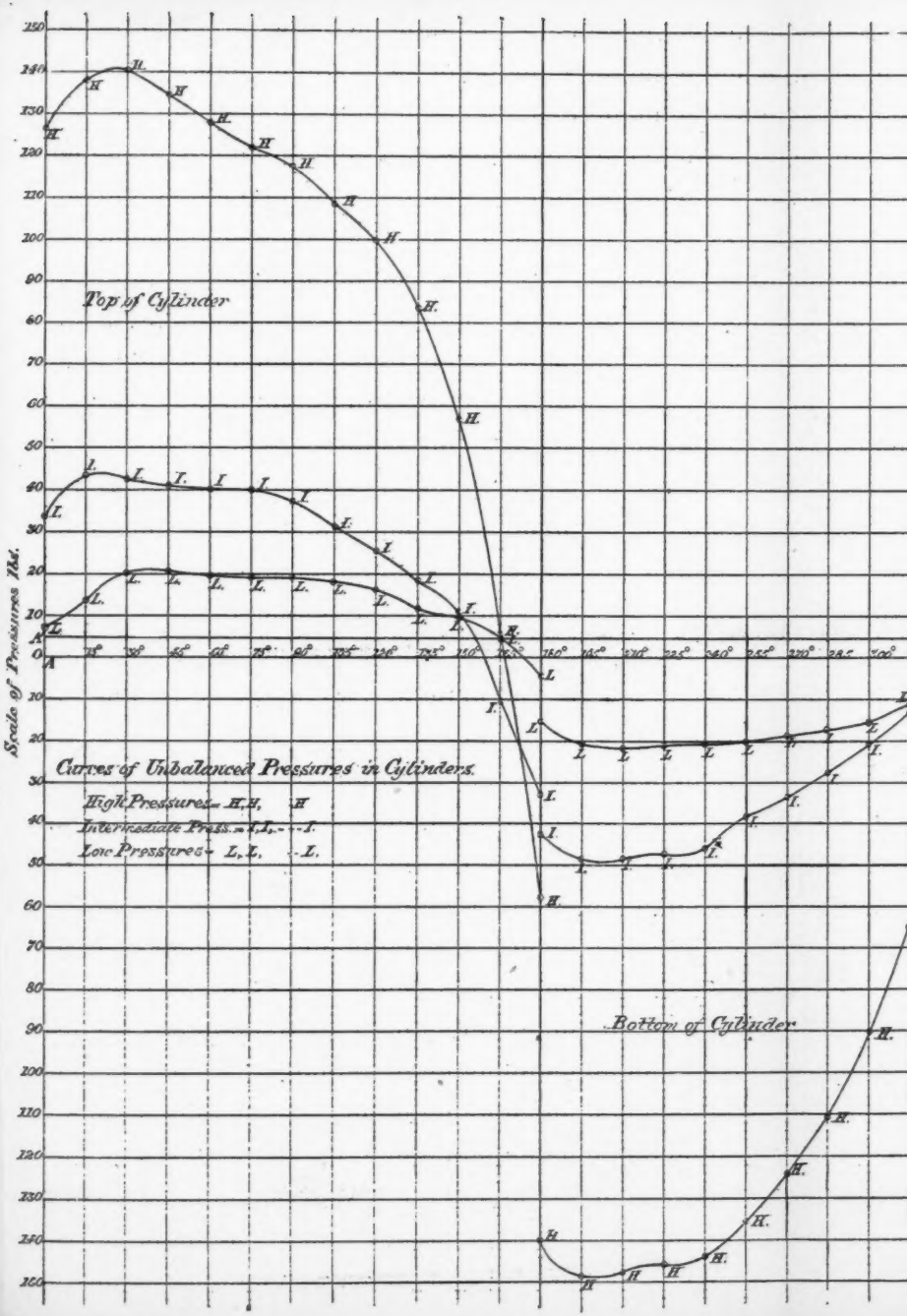
LE III.

			Moments due to inertia.				Combined moments due to pressures and inertia.				
n (arranged in regular cylinders.)		Resultant of moment due to steam.	Multiplier for moment due to inertia.	Moment due to inertia $= WR^2 s^2 \times \omega$.			Combined moment due to steam and inertia (arranged in regular order of cylinders.)				Total moment.
13.	14.	15.	16.	17.	18.	19.	20.	21.	22.	23.	24.
1st L.P.	2d L.P.		ω	H.P.	I.P.	L.P.	H.P.	I.P.	1st L.P.	2d L.P.	
12865.	14430.	27295.	0	0	0	0	0	0	+ 7548.	+ 19747.	27295.
12080.	12630.	40122.	−.00003125	− 12640.	− 13520.	− 8508.	− 2570.	− 476.	+ 13572.	+ 22280.	32806.
10460.	9507.	49997.	−.00005012	− 20270.	− 21680.	− 13650.	− 840.	− 640.	+ 19210.	+ 20457.	38187.
6045.	5319.	52234.	− .00004985	− 20160.	− 21560.	− 13570.	+ 5370.	− 190.	+ 19615.	+ 15091.	39886.
3109.	3180.	54359.	−.00003214	− 13000.	− 13900.	− 8750.	+ 15740.	+ 1930.	+ 16759.	+ 10252.	44681.
1260.	800.	50250.	−.00000548	− 2216.	− 2370.	− 1492.	+ 26774.	+ 3870.	+ 9768.	+ 4462.	44874.
0	0	45230.	+.00001953	+ 7898.	+ 8448.	+ 5317.	+ 35358.	+ 9322.	0	0	44680.
3410.	2923.	44353.	+.00003545	+ 14340.	+ 15330.	+ 9650.	+ 37440.	+ 17290.	− 5098.	− 739.	48893.
9250.	5942	43452.	+.00004022	+ 16270.	+ 17400.	+ 10950.	+ 34260.	+ 24170.	− 4400.	− 1130.	52900.
12910.	8672.	38523.	+.00003589	+ 14520.	+ 15530.	+ 9772.	+ 26280.	+ 26741.	− 660.	− 1100.	51261.
14130.	11060.	30770.	+.00002598	+ 10510.	+ 11240.	+ 7072.	+ 16090.	+ 21680.	+ 5380.	+ 110.	43260.
14820.	12610.	25741.	+.00001345	+ 5439.	+ 5818.	+ 3662.	+ 5916.	+ 11354.	+ 13328.	+ 2960.	33558.
14430.	12865.	27295.	0	0	0	0	0	0	+ 19747.	+ 7548.	27295.
12630.	12080.	39434.	−.00001345	− 5439.	− 5818.	− 3662.	+ 1101.	− 5336.	+ 22280.	+ 13572.	31617.
9507.	10460.	47907.	−.00002598	− 10510.	− 11240.	− 7072.	+ 2370.	− 6620.	+ 20457.	+ 19210.	35417.
5319.	6045.	50324.	−.00003589	− 14520.	− 15530.	− 9772.	+ 4190.	− 1310.	+ 15091.	+ 19615.	37386.
3180.	3109.	53649.	−.00004022	− 16270.	− 17400.	− 10950.	+ 7640.	+ 9550.	+ 10252.	+ 16759.	44201.
800.	1260.	53240.	−.00003545	− 14340.	− 15330.	− 9650.	+ 12250.	+ 22220.	+ 4462.	+ 9768.	48700.
0	0	48190.	−.00001953	− 7898.	− 8448.	− 5317.	+ 17567.	+ 31173.	0	0	48740.
2923.	3410.	47563.	+.00000548	+ 2216.	+ 2370.	+ 1492.	+ 26366.	+ 32410.	− 739.	− 5098.	52939.
5942.	9250.	45772.	+.00003214	+ 13000.	+ 13900.	+ 8750.	+ 31380.	+ 29600.	− 1130.	− 4400.	55450.
8672.	12910.	39522.	+ 00004985	+ 20160.	+ 21560.	+ 13570.	+ 31190.	+ 2440.	− 1100.	− 660.	51870.
11060.	14130.	32053.	+.00005012	+ 20270.	+ 21680.	+ 13650.	+ 24080.	+ 14293.	+ 110.	+ 5380.	43863.
12610.	14820.	25867.	+.00003125	+ 12640.	+ 13520.	+ 8508.	+ 12048.	+ 4847.	+ 2960.	+ 13328.	31183.
.....	.....	.....	0	0	0	0	0	.....	.....	.....	.....

Total = 1,013,142.

Mean $= \frac{1013142}{24} = 42,213.$

Total = 1,013,142.


Scale of Pressures lbs.
Top of Cylinder
Bottom of Cylinder
Curves of Unbalanced Pressures in Cylinders.
High Pressures - H, H, H
Intermediate Press. - I, I, - - I.
Low Pressures - L, L, - L.

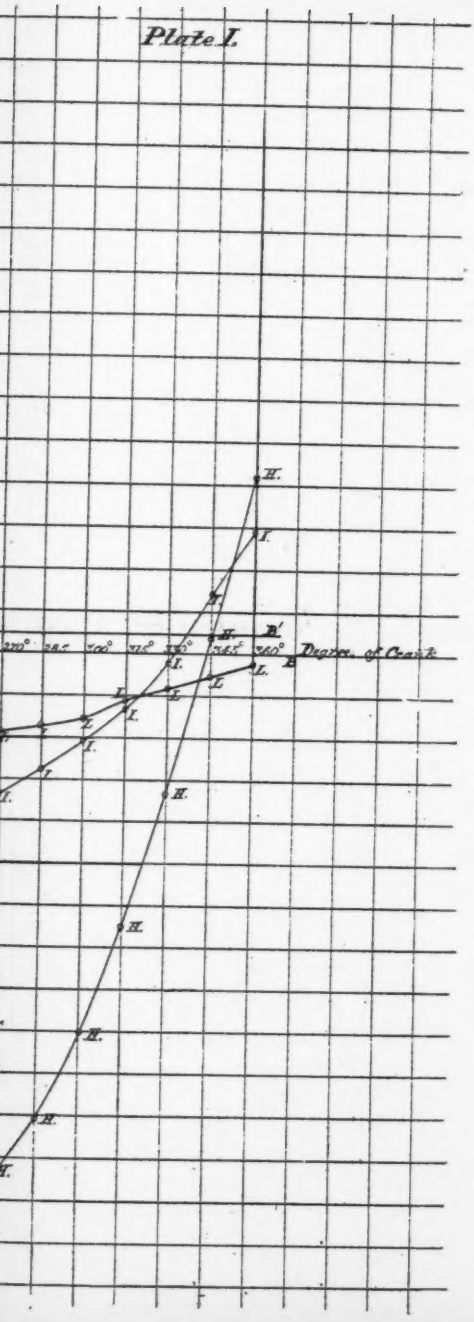
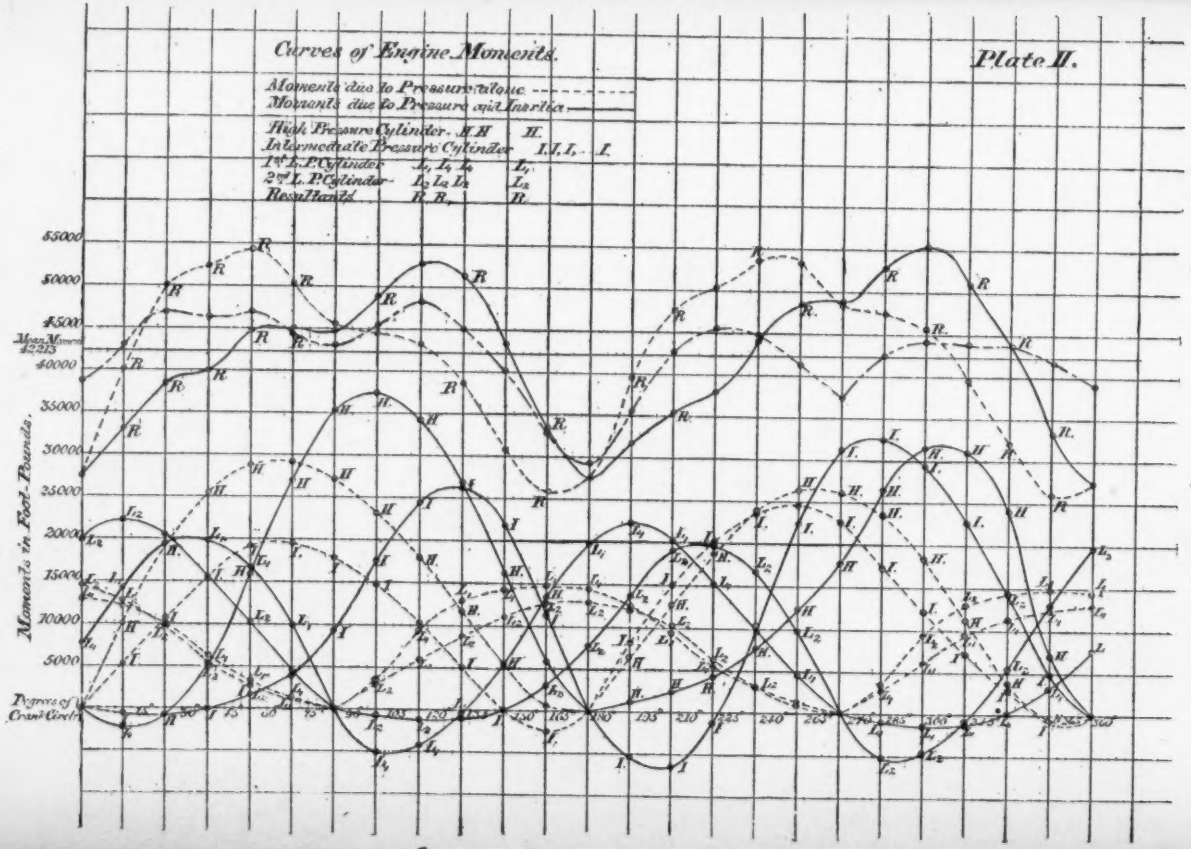


Plate I.
H.
I.
B'
B
Degrees of Crank.


Curves of Engine Moments.
Plate II.
Moments due to Pressure alone
Moments due to Pressure & Inertia
High Pressure Cylinder. H H H.
Intermediate Pressure Cylinder I, I, I, I.
1st L. P. Cylinder L1 L4 L6 L1,
2nd L. P. Cylinder L3 L5 L2 L2
Resultants R R1 R.
Moments in Foot-Pounds.
Mean Moment 42213
Degrees of Crank Circle

NEW POLICE BOAT FOR NEW YORK CITY.

By H. deB. Parsons, Esq., Associate.

Believing it a wise policy to record actual performances, the author offers the following with the hope that it may assist others in designing a similar vessel.

In the Spring of 1893, the Police Commissioners decided to construct a new police boat to replace the old-fashioned side-wheel steamer then in service. The author was retained as Consulting Engineer and Inspector for the work. It was decided to build a steel twin-screw vessel to be called *Patrol*, and the contract was awarded to the marine department of the Maryland Steel Company, Sparrow's Point, Md. The contractors were to complete the vessel and deliver it in New York City within six months. This was accomplished, with two days to spare.

The dimensions of the boat were:

Length over all, feet and inches	143–8
Length between perpendiculars, feet	135
Beam, feet	23
Depth, moulded, feet	10–6
Draught, loaded, bow, feet	6
Draught, loaded, stern, feet	8–6

The stem is straight and the stern has a round overhang. In general appearance the boat resembles a large ocean tug with two pole masts, having a long deck house which is utilized for saloons, state rooms, engine and galley.

On the lower deck forward is the forecastle with accommodations for ten men. Aft of this is the main dining room and two large staterooms, then a cross coal bunker, and the boiler and engine rooms respectively. Abaft the engine room there are six state rooms. The overhang of the stern is used for a store room for general supplies.

The designing weights and data were:

Displacement, tons	274.00
Steel in hull, tons	88.50
Machinery, including boiler settings, tons	94.00
Carpenter and joiner work, tons	38.00
Outfit and stores, tons	10.00
Coal, tons	20.00
Cement and paint, tons	8.50
Fittings and rigging, tons	15.00
Moment to alter trim one inch, tons	36.45
Displacement per inch at L.W.L., tons	5.18
Transverse metacenter above C.G., feet	3.90
Longitudinal metacenter above C.G., feet	223.00
Cylindrical coefficient of displacement	0.649
Block coefficient of displacement	0.426
Area load water plane, square feet	2,174.6
Cofficient load water plane	0.700
Area midship section, square feet	109.5
Coefficient of midship section	0.656

The frames were steel angles 3 by $2\frac{1}{2}$ inches, spaced 21 inches, and the reverse frames were $2\frac{1}{2}$ by $2\frac{1}{2}$ inches. The keel was of the bar type, of steel, 6 by $1\frac{1}{2}$ inches. The stem was of the same size as the keel, and the stern post was cast steel, 6 inches by 3 inches at top, tapering to $1\frac{1}{2}$ inches at heel.

The motive power consisted of two compound direct-acting inverted engines. The high-pressure cylinders were 13 inches in diameter, and the low pressures 24 inches, both with 18-inch stroke, with cranks at 90 degrees. All the cylinders were unjacketed. The high-pressure cylinder had a solid piston valve, and the low a double-ported slide valve. The engines were reversed by steam, and the valves were direct-driven on the Stephenson link, double-bar type. The air, sanitary and boiler feed pumps were connected to the main engine low-pressure crossheads. Independent centrifugal pumps, having 4-inch suction and discharge, were used for circulating. Each condenser contained 390 square feet of cooling surface, and formed part of the back framing for the engine. The condenser tubes were of brass, tinned inside and out, $\frac{3}{4}$-inch diameter, spaced $1\frac{1}{8}$ inches. The shafts were steel, 5 inches inboard and $5\frac{1}{2}$ inches

outboard. There were five thrust collars, each 1¼ inches thick by 2¼ inches high. The cranks and crank pins were cast steel, in one piece, and were bolted to flanges forged on the shafts. The propellers were cast iron, containing a large percentage of steel scrap, 6 feet diameter by 9 feet pitch, and weighed 1,050 pounds each. The weight of each engine on floor of erecting shop was approximately 21,000 pounds.

The steam pressure was 120 pounds and was supplied by two boilers of the "Admiralty" or gunboat type, 7 feet 9 inches diameter by 17 feet 6 inches long. Each boiler contained two corrugated furnaces, 36 inches inside diameter, and 78 3-inch tubes and 70 3½-inch tubes. The lower tubes were made larger to prevent their being choked by soot, which is apt to occur with this type of boiler. The grate surface for both boilers was 74 square feet, and the total heating surface 2,130 square feet. The steam space in both boilers was 285.6 cubic feet.

Each boiler weighed empty (estimated) 28,020 pounds, and with water ready for steaming 54,270 pounds.

There were the usual number of auxiliary pumps for bilge and boiler feeds, as well as a steam syphon, and these were connected by manifolds so as to draw from sea, bilge, main tanks or fore and aft peak tanks. In addition, there was a Worthington fire and wrecking piston pump with central bottom suction, duplex, 14-inch steam, 9¼-inch water, by 10-inch stroke, capable of discharging over 1,000 gallons per minute. This was arranged for four hose connections on the upper deck, and for two on the main deck.

The vessel was lighted throughout with electricity in addition to the usual complement of oil lamps. On the foremast was a Thomson-Houston search light of 4,000 candle power, arranged to be operated from inside the pilot house. The side light, mast head and anchor lights were arranged for both electricity and oil.

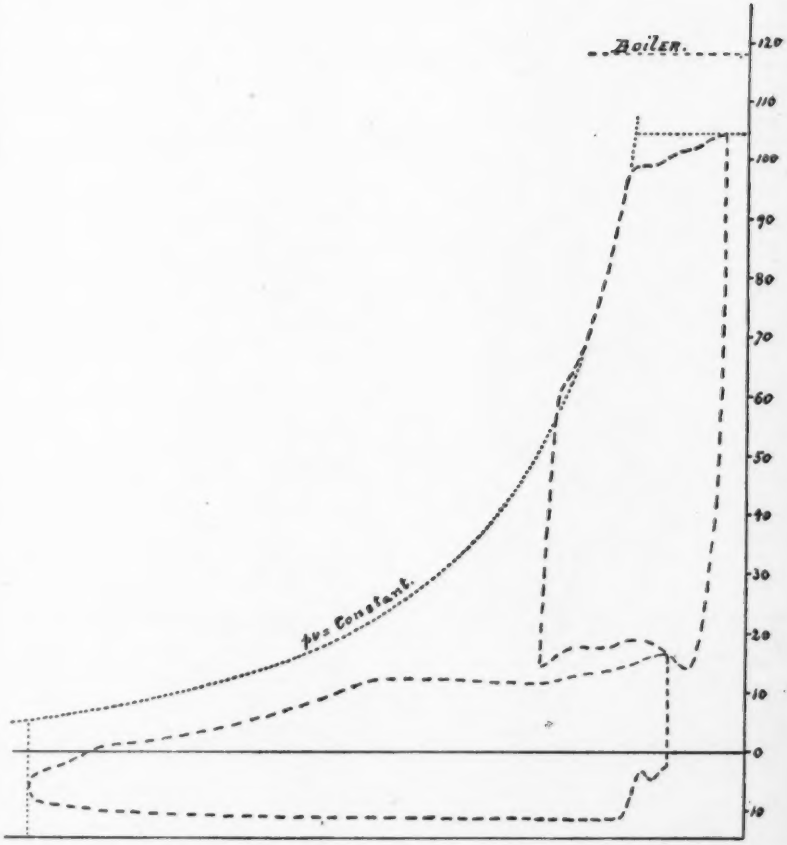
Besides the hand steering gear, there was a Manton steam steerer, so connected that the change from hand to steam could be made quickly while the vessel was under way.

The steam trial took place on January 11, 1894, and lasted four hours. During one hour indicator cards were taken, from which the horse power was computed. The contractors guaranteed a minimum of 500 I.H.P. On account of foggy weather, it was impossible to accurately gauge the speed, but it was approximately 16 statute miles per hour.

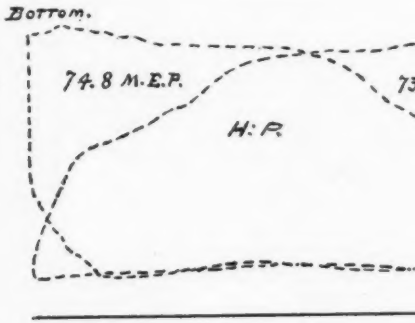
Average cards from both engines are shown in the accompanying drawing, and also a combined diagram.

The results of the trial were as follows:

Speed, approximate, miles			16
Coal,			soft, of good quality.
	Starboard.		*Port.*
I.H.P. greatest	320.02		318.21
least	275.70		273.93
average	298.46		297.26
collective		595.72	
Revolutions, average	185.55		185.66
Steam pressure in receiver, lbs	15.00		15.33
Vacuum, inches	25.50		26.75
Combined diagram—ratio of theoretical to actual area, per cent		61.93	
Height of stack above grate, feet		36.00	
Area of stack, square feet		12.60	
Area through tubes, square feet		14.00	
Area grate surface, square feet		74.00	
Ratio grate surface to area tubes		1 to 5.04	
Ratio grate surface to area stack		1 to 5.87	

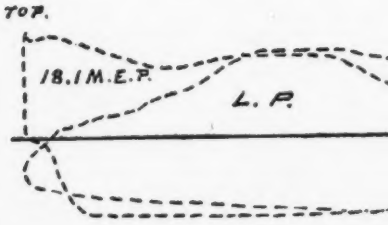

Boiler.
120
110
100
90
80
70
60
50
40
30
20
10
0
10
$p_0 v$ Constant.

PORT.



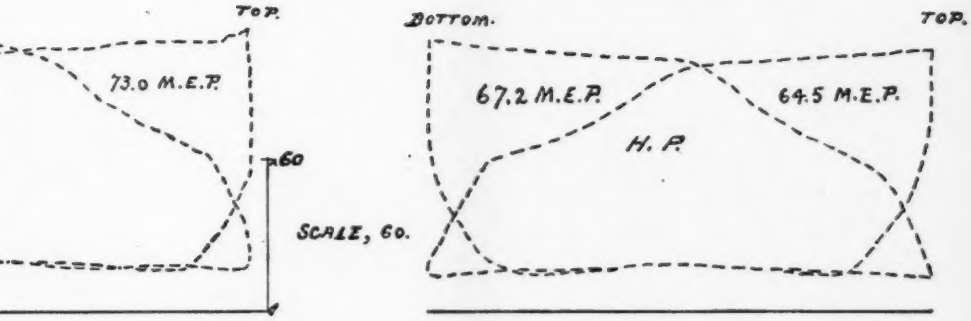
REC. P

VAC. I



RT.

STARB'D.

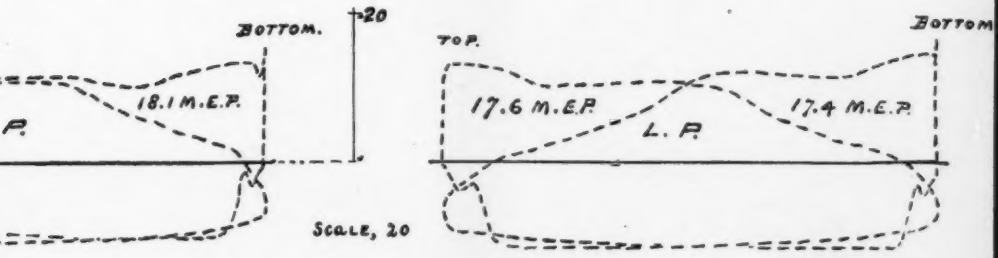


BOILER PRESS: 118. LBS. PER SQ. IN.

REC. PRESS:- 15 LBS. PORT & STARB'D. ENGINES.

VAC. IN INS: = 25 PORT. - 26 STARBOARD.

REVOLUTIONS · 178. PER MIN.



CIRCULATION IN THE "THORNYCROFT" WATER-TUBE BOILER.

By J. I. Thornycroft, Esq., F. R. S.

[Read at the Thirty-fifth Session of the Institution of Naval Architects, March 15th, 1894.]

Since I read a paper on "Water-tube Boilers for War Ships," in 1889, such a change in the general feeling regarding them and their actual standing has taken place that it is unnecessary for me now to point out at any length the decline and failure of shell boilers *for fast vessels*, and the growth and success of the water tube, but rather to describe my most recent practice, and give you the results of my experiments and experiences since my paper before the Institution five years ago.

It was considered then that possibly boilers such as I had made might be suitable for small torpedo boats, but that success in such an extreme case did not at all warrant their adoption in larger vessels. However, in the last two years the third-class cruiser *Geiser*, and the fast torpedo gunboat *Speedy*, have each been fitted with eight water-tube boilers of my construction; and now the Danish authorities are so well pleased with the performance of the *Geiser* that they have decided to fit the armor-clad *Skjold* with similar boilers. Further, in Germany, a new ship of the *Siegfried* class of 3,000 tons is also to be fitted with this type.

In the case of the *Geiser* an equal power was given to that obtained in the sister ship, *Hecla*, with locomotive boilers, but with the reduction of weight of some sixty tons. I wish to call your attention particularly to what my firm undertook to do in the case of the *Speedy*. One thousand more horse power was guaranteed than with the locomotive boiler in sister vessels, with a considerably less weight of boilers.

I would beg to call your attention to Fig. 1, which gives a comparison of locomotive boilers and the *Speedy's* water-tube boilers. First, you will see that we have the same length in the ship in each case, and in each case two stokeholds; but in the *Speedy* four boilers in each stokehold instead of two.

In the table you will see the comparison of heating surfaces, grates, &c. It has been remarked that the amount of heating surface provided is much greater per unit of power than is usually employed with locomotive boilers. This, however, is not a necessary condition, as it has been shown that water-tube boilers are capable of being forced to do equal or more work with equal surface; and the large amount of heating surface in the *Speedy* must be looked upon, not as a necessity, but as a means of providing greater economy of fuel. Coal consumption experiments in the *Speedy* have not, at present, been made, but in the *Geiser* careful experiments over a considerable period show that the coal consumption, even at full power, is considerably less than two pounds per I.H.P.

PARTICULARS OF LOCOMOTIVE AND THORNYCROFT BOILERS IN VESSELS OF THE *SPEEDY* TYPE.

	Locomotive boilers.	*Thornycroft water-tube boilers.*
Total indicated horse power	3,500	4,500
Number of boilers per ship	4	8
Area of grate, per boiler, square feet	45½	25½
Total area of grate, per ship, square feet	182	204
Heating surface, per boiler, of firebox, square feet	272¾	64.6
Total heating surface, per boiler, square feet	1,597	1,840
Total heating surface, per ship, square feet	6,388	14,720
Total weight of boilers and mountings, dry, tons	82	71.52
Total weight of water in boilers, tons	30	12.78
Total weight of boilers and water, per ship, tons	112	84.30

During the trials of the *Speedy* we had some difficulty with priming, which had not been expected, and this was found to be entirely due to the quality of the water, which was the Sheerness Dockyard supply, and which has been found so far capable of causing trouble that no one of experience will use it willingly. And in the case of the *Speedy* the difficulty was further increased

by a large addition of estuary water, caused by a leak in the condenser. I have since made experiments on the properties of these waters, which I shall refer to later on.

When working at the highest power attained the boilers were only forced to a very limited extent, the mean air pressure being 1.7 inches, and, even with this air pressure, had the Navy stokers had more practice with the particular form of grates, a higher result could have been obtained. The only point which presented any difficulty in these trials was that of feeding the boilers with sufficient regularity. The *Speedy* has two Weir pumps, which are arranged to deliver into a main feed pipe common to all the boilers, in which the water was in excess of the boiler pressure by some 50 pounds. On each boiler there was a check valve, by which the feed was controlled. This system, which is perfectly efficient for boilers holding large volumes of water, seems to require some modification in the case of water-tube boilers, where the water contained is relatively very small for the rate of evaporation, and I have come to the conclusion that some form of automatic control is desirable. In the Belleville boiler this has already been adopted with success. With the view to lengthen the life of the tubes, they were galvanized both inside and out. The zinc on the outside is, I consider, more useful than the inner coating, which, being exposed to impure water at a high temperature, soon disappears, and is, therefore, only a temporary protection.

In opening up boilers and steam pipes we have frequently experienced the presence of explosive gases in a most unpleasant manner. It seems to me possible that the very large surface of zinc must be the cause of the generation of these gases. This theory is supported by the fact that we have only had this experience since the tubes have been galvanized, and also by the disappearance of the zinc. In one case, during the severe frost this winter, a torpedo boat had been lying with banked fires for five days. A leak was discovered in the exhaust pipe of the steering engine, and the presence of a combustible gas was accidentally made evident by its ignition, when it burned for between three and four minutes, the flame being about three feet long, burning

at first with a blue flame, and later with a luminous flame like coal gas.

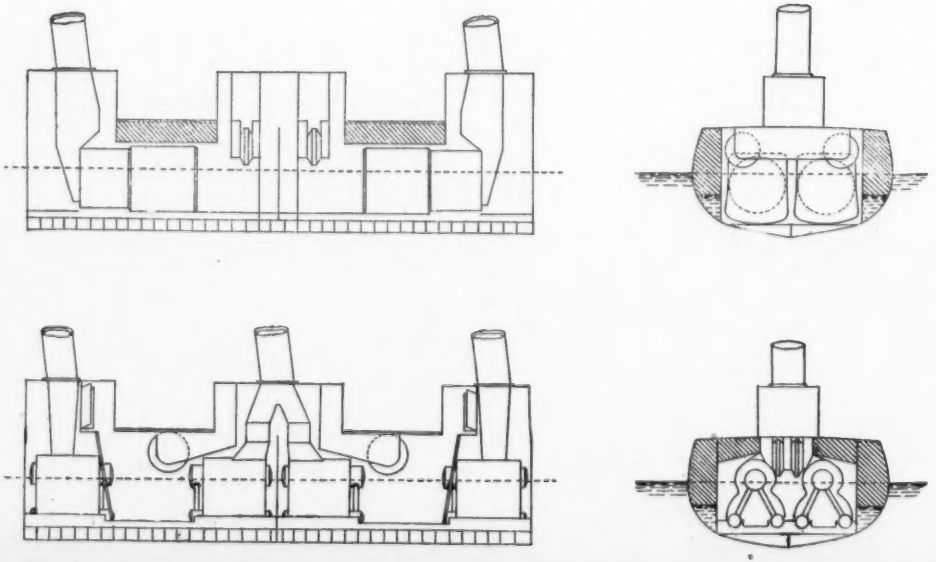
On studying the papers of Mr. Milton at Cardiff, and Mr. Ward at the Engineering Congress of Chicago, I find what other engineers have been doing; and while I have pleasure in acknowledging the honor many have done me in following the principles of my construction, I must call attention to the fact that I am of opinion that many of the departures that have been made will lead to inefficiency and may be sources of danger.

I have recently made experiments on the relative circulation of boilers when the generating tubes deliver above the water in the separator and below it, and I have found that, in the case where they deliver above, the circulation is rather more than double when they deliver below. The method of measurement I adopted was to put a rectangular notch, similar to those usually employed in gauging small streams, across the separator, so that all the water that went down the downtakes had to pass over it, and then observed the flow over the notch through a glass window in the end of a boiler, and thus, knowing the size of the stream, was able to calculate the circulation. I found that, in the case of the above water delivery, the circulation was 105 times the feed; that is to say, for every pound of steam brought up by the generating tubes, 105 pounds of water are also passed through them. The boiler experimented upon with the drowned tubes was similar, as far as possible, to the other, which was of my usual construction, so that the reduction in circulation can only be attributed to the upper ends of the tubes being immersed.

At a meeting of the Civil Engineers in 1889, I described at length the necessary conditions for efficient circulation, and ventured an opinion that it, together with the proper heating of the feed water before its arrival at the heating surface, would retard the formation of scale; and I am now able to state that we have had no trouble in any of our boilers from this cause. It is, therefore, apparent the large reduction in circulation caused by immersing the tops of the generating tubes will detract from their

DIAGRAM SHEWING SPACE OCCUPIED BY LOCOMOTIVE & THORNYCROFT WATER-TUBE BOILERS IN VESSELS OF THE 'SPEEDY' TYPE.

FIG 1.



power of keeping clean, which is so essential to boilers having tubes of a small diameter.

It has been argued that it would be advantageous to have the tubes drowned, because it has been thought that then brass or copper might be used instead of steel; but my experiments show that the proportion of steam to water is greater in the drowned tubes; therefore they have less capacity to take up heat from the furnace gases without damage.

I have made experiments which proved that working with water entirely free from oil, very severe working (20 pounds of water per square foot H.S.), the tubes could not be overheated; but the addition of a small quantity of the greasy deposit taken from a boiler working with condensing engines was sufficient to cause the destruction of the tubes by overheating without any increase in the rate of working. I further experimented with grease taken from the inside of a boiler to ascertain the temperature of its evaporation, and found that this was so high that when heated on a piece of brass tube it continued to give off vapor after the critical temperature at which brass loses most of its tenacity had been arrived at. From this experiment the effect of a coating of this deposit can be easily surmised. Under the rapid transmission of heat, gas would continue to be liberated from the surface, which would increase in temperature, owing to the non-conducting power of the film of gas which necessarily precludes the contact of water with the tube surface, the water being only enabled to preserve the tubes because its boiling point is lower than that of grease. Consideration of these facts seems to me to prove that for a high rate of working in boilers where grease may be present in the feed water brass tubes are quite unsuitable; copper may, perhaps, be used, as the temperature at which its tenacity is greatly reduced is much higher. M. Du Temple found that copper tubes lose their strength in time, but the relatively small diameter of his tubes to their length seems to be an element which might cause this trouble, and a further unfavorable condition is the immersion of their upper ends.

In the experimental boilers before referred to, and in which glass ends to the separators enable internal working of the

boilers to be observed, I made experiments on priming, and the phenomenon which takes place seems both important and interesting. Waters which cause priming on boiling produce foam consisting of a mass of bubbles of various sizes. Water which is very bad produces bubbles so durable as to remain a considerable time without breaking, and by them the steam space of a boiler may be entirely filled; and so soon as this takes place, instead of simply steam leaving the boiler, the discharge consists of foam, which is broken up in its rapid motion along the steam pipes. This is a great contrast to what is seen when pure water is evaporated. In this case steam emerging from the surface of the water retains no film of liquid for sufficient time to be seen; this is what might be expected, as pure water is known to be incapable of forming bubbles.

The effect of this is very marked in the discharge from the tubes, both in delivering above and below water. In the former case, with pure water, the discharge of steam and water is periodic. The water and steam are discharged quite separately, but with a priming water the discharge is a steady one, consisting of a mixture of steam and water in the form of very wet foam. In the case where the tubes discharge below water, the flow appears to be unsteady in both cases; when the water is pure, the water and steam are discharged together, as in the above water delivery, then at the pause in the flow the water runs back into the tube on to the ascending column, so that there is to some extent an alternating flow in the upper ends of the tubes. This occurs in the same way with a priming water, only to a more marked extent, columns of foam being thrown the full height of the separator. In this way tubes delivering below water are capable of causing a priming effect, when tubes delivering above will not do so.

The impurities in water which may cause priming are not easy to determine. It is well known that waters which do not prime by themselves, when mixed do so in a most astonishing manner; for example, either salt water or pure water may work in a boiler without priming, but by mixing the two very bad priming is started. I have tried water taken from the Thames

at the Nore, and well water from Sheerness; both these are good samples of priming waters, but when mixed in about equal proportions produce such a priming mixture that the very slowest rate of evaporation causes violent foaming. As this mixture corresponds with the water used during part of the *Speedy's* trials, the priming then experienced is now explained.

Returning to the question of circulation. It has generally been admitted that it is necessary to provide tubes other than the steam generators for conveying the water from the separator to their lower ends. If this is not done, it is evident that some of the generating tubes must serve this purpose; or if this is not the case, the motion of water in the tubes will not be a circulation, but only motion alternating in direction, which is not favorable for preventing the formation of scale, or providing the necessary supply of water to prevent their over-heating.

At first sight, it would appear that in my improved type of boiler as fitted in H.M.S. *Daring* [see page 156 of last number of the JOURNAL], the heat to which the down tubes are subjected might be a source of impaired circulation; but, if the question is examined, it will be found that owing to the fact that care is taken to mix the feed water with the large volume of water in circulation, and that these tubes are only exposed to the cooler gases, the heat required to bring the descending columns to the boiling point would be more than could be obtained in this way, while at the same time all heat added below the boiling point must be looked upon as a gain in efficiency, this arrangement at the same time making the boiler more compact, and enabling a larger fire grate area to be obtained in a given boiler-room. A gain in this direction is further increased by putting boilers side by side, and leaving out what in one alone would be the external fire-box walls.

The fact that, in such an arrangement, the boilers form as it were one continuous mass has many advantages, including great simplicity of the casings, together with the reduction of their external area to perhaps a minimum. At the same time the exit for the gases from the boilers being along the center of the ship, they require very simple uptakes to the funnels.

As each fire is common to two boilers, one boiler cannot be put out of action without another boiler losing half its fire, but this boiler would still do half duty, as half its surface would be in no way affected. So I do not consider this any objection to the arrangement. The very large volume of space above the fire in these boilers is very advantageous to ensure complete combustion of the furnace gases. In boilers of the *Speedy* type this space had already been increased far beyond that usually adopted in other boilers, but in this type the fire box is larger still, and I think there is no doubt that an advantage is thereby gained.

In some boilers where this space is limited, special means have to be provided to ensure proper combustion before the gases come in contact with the heating surface.

WATER-TUBE BOILERS.

By J. T. Milton, Esq.

[Read at the Thirty-fifth Session of the Institution of Naval Architects, March, 15, 1894.]

At the meeting of the Institution held at Cardiff, in July last, I had the honor to read a paper "On the present Position of Water-Tube Boilers for Marine Purposes." Since that date some more experience has been had with these boilers, and, as the matter is one in which all marine engineers take much interest, and in which many of them are now exercising their ingenuity and inventive talent, I have thought that another paper, giving further information of some types of boiler now being made, and inviting criticisms of points of detail, both of principle and of construction, would meet with the approval of the Institution, and would be of service at the present time.

Since writing the major portion of this paper I have received the proceedings of the International Engineering Congress, division of Marine Engineering, held at Chicago in July and August last. These contain an excellent paper on tubulous or coil boilers by Mr. C. Ward, the originator and maker of a type of water-tube boiler successfully fitted on board the U. S. S. *Monterey*. The boilers described in this paper are (1) Ward's boiler, consisting mainly of nests of tubes one above another, bent into the form of concentric circles, each tube being, however, attached at one end to a descending header, and at the other to an ascending header leading into the steam spaces; (2) the Cowles boiler; and (3) Mosher's boilers, which may both be considered to be modifications of the Thornycroft boiler; (4) the Towne boiler, consisting of water-slab casings, with two sets of inclined water tubes crossing each other, and a steam drum; (5)

the new Herreshoff; and (6) the Roberts boilers, consisting of rectangular casings surrounding the fire. The paper further treats of the Belleville, the Thornycroft, and the Yarrow boilers, and is well worth careful study by all interested in water-tube boilers.

Since reading the paper at Cardiff, the Babcock and Wilcox boiler therein described has been in continuous work at sea on the steamship *Nero*, and up to the present time has given satisfaction. The Thornycroft boiler, also, as fitted in H. M. S. *Speedy*, has been successfully tried, and has fully answered expectations.

At the present time, a large number of water-tube boilers are being fitted in vessels for the Navy. The Belleville boiler is being fitted in H. M. S. *Sharpshooter* and in two new armorclads. Another gunboat is being fitted with the Du Temple boiler, while the large number of new torpedo-boat destroyers being built are nearly all being fitted with water-tube boilers of various types, including the Thornycroft, Normand, Yarrow and Du Temple, previously described, and other types by Messrs. J. S. White, A. Blechynden and others. It must be admitted that, but for water-tube boilers, the very high speed of these vessels could not be obtained; so that if these boilers had not been available, the vessels themselves would probably have had no existence.

Turning to the merchant navy, Messrs. Fleming and Ferguson are now constructing a pair of water-tube boilers for a vessel building by them, intended to be classed by Lloyd's Register. These boilers are shown on Plate I. Each consists of a steam and water chest, 6 feet in diameter, and two water chambers, 3 feet in diameter, extending the whole length of the boiler, and connected by numerous bent tubes. These chambers are connected outside the boiler casing by external circulating pipes of large diameter. The fire grate extends practically over the whole space below the water chambers, and the products of combustion rise up amongst the tubes on the way to the chimney.

The tubes—which are of iron, lap-welded, $2\frac{1}{2}$ inches in diameter—are expanded in the chests and are afterwards beaded over. The chambers are built into the frame work of the boiler casing

before the tubes are fitted, so that the tubes have no strain on them due to the weight of steam chest, &c.

It will be noticed that the tubes are all made with the upper end $2\frac{9}{16}$ inches in diameter, and their length is such that they can all be put in place from the inside of the steam chest. Any one tube can be cut out and replaced without disturbing the neighboring tubes. Although the tubes are curved, owing to their comparatively large diameter, a light held at one end may easily be seen to shine through the other end if they are clear.

A boiler of this general type has been at work on shore in Messrs. Fleming and Ferguson's works for some time, and has given satisfaction, both as regards evaporative efficiency and also in giving no trouble whatever.

The following table gives some particulars supplied by Messrs. Fleming & Ferguson, of the weights, surfaces, &c., of these various types of boilers :

TABLE I.

Type of boiler.	Working pressure, lbs. per square inch.	Heating surface in square feet.	Grate surface in square feet.	Total weight of boiler and contained water (cold), with casings, and all mountings and fittings, except chimney.	Total weight per square foot of heating surface.
				Tons.	*Lbs.*
Navy type, one furnace	200	1,150	30	21	41
Two boilers for steamer, where height and width are kept a minimum, both	200	1,450*	44*	34*	52
Single-ended boiler, with two furnaces	200	1,150	48	23	45
Double-ended boiler, four furnaces	200	2,200	88	38	39
Single-ended boiler, with three furnaces	200	1,550	60	31	45
Double-ended boiler, with six furnaces	200	3,400	110	55	36
Double-ended boiler, with six furnaces	200	4,800	138	75	35
Boiler being made, as per Plate I.	220	1,450	50	30	46
Large boiler, with four furnaces fired cross-ways	250	5,000	165	110	49

* Two boilers.

Messrs. Anderson & Lyall, of Glasgow, have made a boiler on the plan illustrated in plate II, but this type has not yet been tried at sea. The makers are of opinion that in order to abstract as much heat as possible from the products of combustion, they must be subdivided into a number of small streams, each of which must be forced to act upon a considerable extent of surface, and, to insure this, they have retained some of the heating surface in the old form of smoke tubes.

In this boiler the water-tubes are not spaced close together, and a large space or combustion chamber is formed between them and the barrel containing the smoke tubes. These features of the design are intended to provide for the proper combustion of the furnace gases; and in actual use considerable combustion and much evolution of flame takes place above the water tubes.

The design illustrated shows water sides to the fire space, but the makers have proposed plans for forming these sides with iron casings, protected from the intensity of the fire by water tubes nearly in contact with one another, as is done in several other types of boiler.

Messrs. Anderson & Lyall inform me that the total weight of boiler and all accessories, including chimney and water, of an installation they have under consideration, is 40 tons 3 cwt. The boiler will have a grate surface of 69 sq. ft., and a total heating surface of 2,340 sq. ft. This weight is equivalent to about 38 lbs. per sq. ft. of heating surface. The working pressure will be 180 lbs.

The boiler appears to have a very high evaporative efficiency. During a five-hours' trial of January 31, 1894, with the shop boiler made on this plan, the evaporation was equivalent to 12.05 lbs. from and at 212°, the rate of combustion being 18.29 lbs. per sq. ft. of grate. The coals used were Welsh, samples of which were kindly tested by Mr. C. J. Wilson, and found to have a calorific value of 14,467 calories, equivalent to an evaporation of 14.97 lbs. of water from and at 212° Fahr. The steam generated was practically dry, the wetness found by the most approved method being less than one per cent. It may be stated that the trial

PLATE 1. FLEMING & FERGUSON'S BOILER

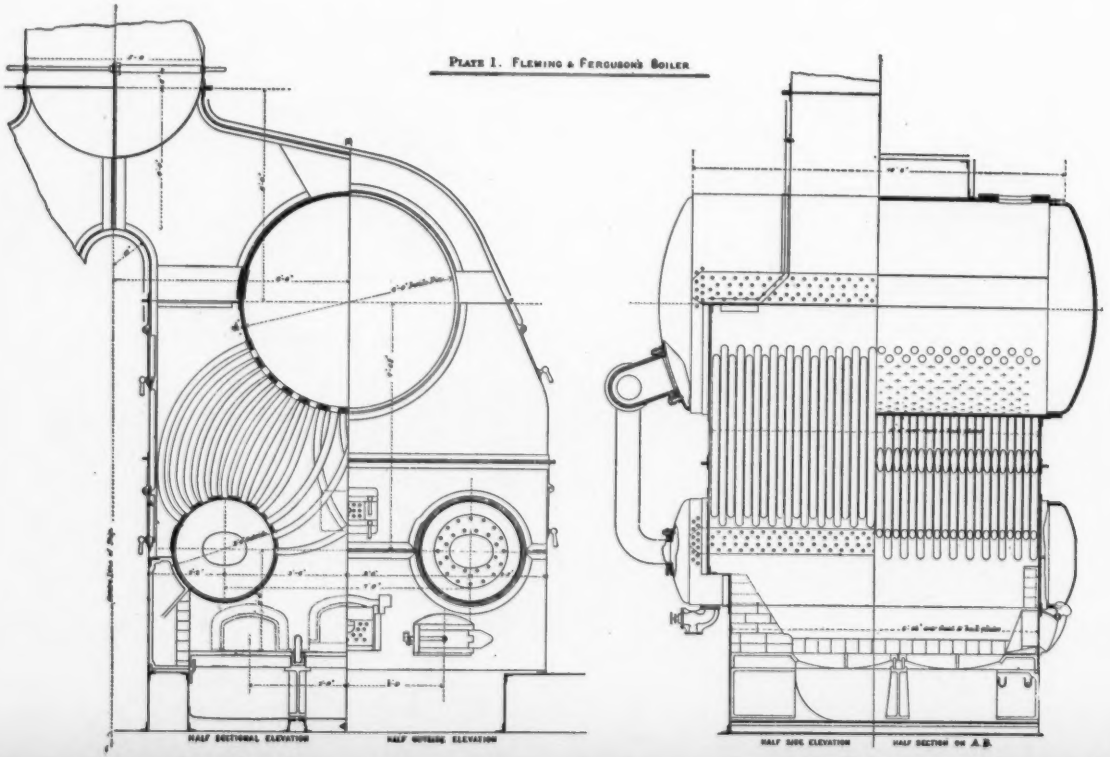
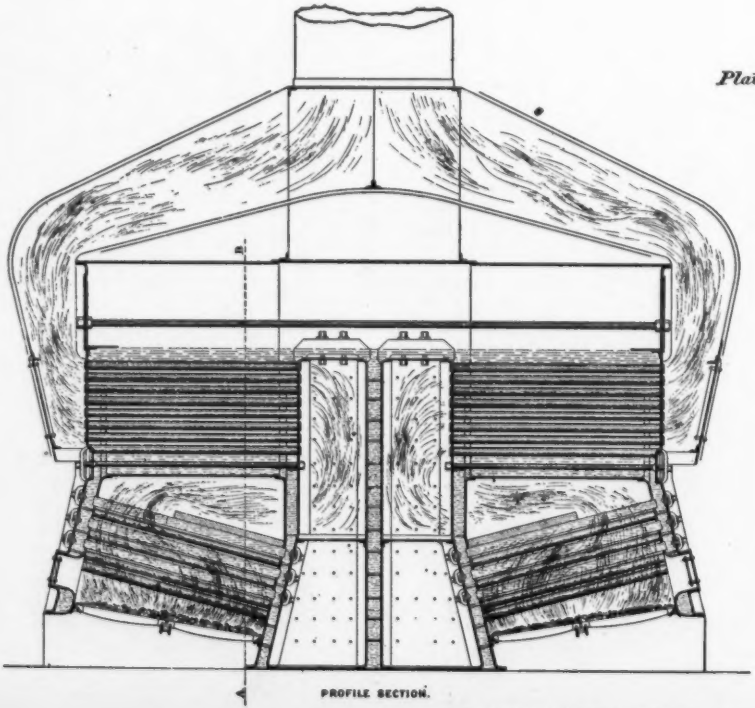
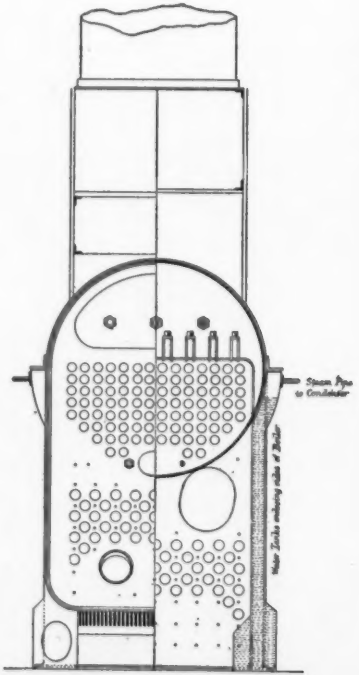


Plate II.



PROFILE SECTION.



FRONT ELEVATION. ELEVATION SECTION AT A B

—— LYALL'S PATENT BOILER. ——

was controlled by my colleague, Mr. Stromeyer, and was therefore, conducted entirely independently of the makers.

A form of boiler used on the Continent is that known as the Dürr boiler. Those intended for marine purposes are very similar in design to the land boilers made by the same firm. One of the marine type is shown in Plate III.

The chief parts of these boilers are:

(1) A water chamber extending over the front of the boiler divided into two parts by a diaphragm plate, which is made in portable pieces, each being secured by nuts threaded on the screw stays.

(2) A number of slanting rows of tubes communicating at their upper ends with this water chamber, the lower ends being closed.

(3) One or more steam receivers placed over the water tubes, connected at the front end to the water chamber.

(4) A nest of superheater or drying tubes, placed over or between the steam receivers.

The water tubes are made at their front ends with rings welded on and turned conically, the conical portions fitting into the milled holes in the back plate of the water chamber, without requiring any expanding, rolling or jointing of any kind. As the tubes are placed at an inclination, while the water chamber is vertical, the tube ends have to be turned in a special manner to fit at the proper angle. The diameter of the tubes at the rear ends is somewhat reduced; these ends are closed by an end plate fitting with a conical joint kept in place by one bolt, with cross-bar, straps, &c. The tube ends are carried on an iron plate forming part of the framework of the boiler, protected with bricks, and the tubes are perfectly free to expand or contract.

Circulation is obtained by means of internal concentric tubes fixed to the diaphragm plate, and communicating with the front part of the water chamber. These inner tubes reach nearly to the end of the water tubes.

The water level of the boiler in ordinary working is about the center of the steam receivers. The water passes from these receivers down the front part of the water chamber, then through

the inner tubes into the outer tubes, where part of it is evaporated. The steam and water then pass out of these tubes into the rear part of the water chamber, whence they are led into the receivers.

The water tubes at the sides are placed as near each other as possible, to prevent loss of heat by radiation. This is effected by bending them alternately to the right and left.

A hole is provided in the front plate opposite each water tube, to enable it to be drawn out or replaced. The holes in the outer plate are closed by hollow caps with conical fitting portions placed from the inside, and, like the tube ends, these caps fit tight without requiring any rolling or jointing of any kind. The taper ends of the tubes, and also of the caps, are untooled at the extreme ends; these portions, therefore, are of slightly larger diameter, the collar forming a stop, which is a safeguard against their being blown out from any cause. The tubes are cleaned on the outside by a steam jet.

Baffle plates are fitted between the tubes to ensure a proper circulation of the furnace gases amongst the tubes.

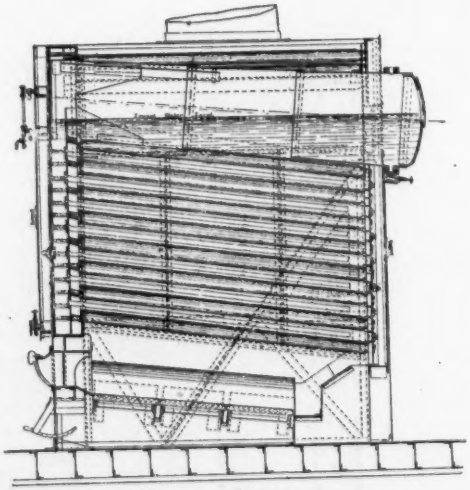
The superheater consists of concentric tubes similar to the water tubes, and the steam circulates through them in the same way, first passing through the inner tube, and then through the annular space between the tubes, where it is dried, or, possibly, superheated.

The makers of these boilers, Messrs. Dürr & Co., of Ratingen, are represented for this country by Messrs. Van Rietschoten & Houwens, of Rotterdam, who have been good enough to furnish me with the particulars of these boilers. Although making principally land boilers, of which they have made 944, having a collective heating surface of over 1,000,000 square feet, during the last six years, they have made several marine installations—among others, two being for the German Navy. These boilers are also said to be giving perfect satisfaction in large paddle towing steamers working on the Rhine. Messrs. Rietschoten & Houwens inform me also that the German naval authorities have made experiments with these boilers, forcing them to a consumption of coal at the rate of 70 lbs. per square foot of grate per hour without developing any defects whatever.

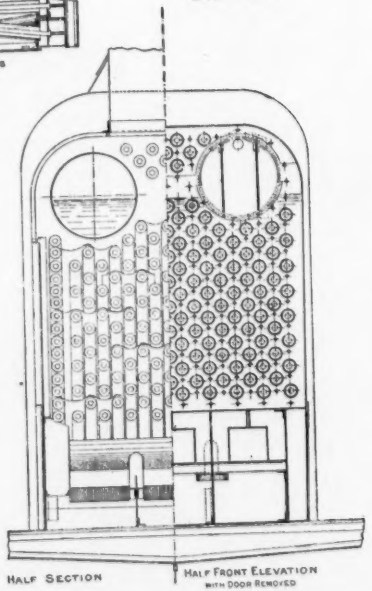
DÜRR BOILER

Plate III.

PLAN SHEWING BENDING OF SIDE TUBES



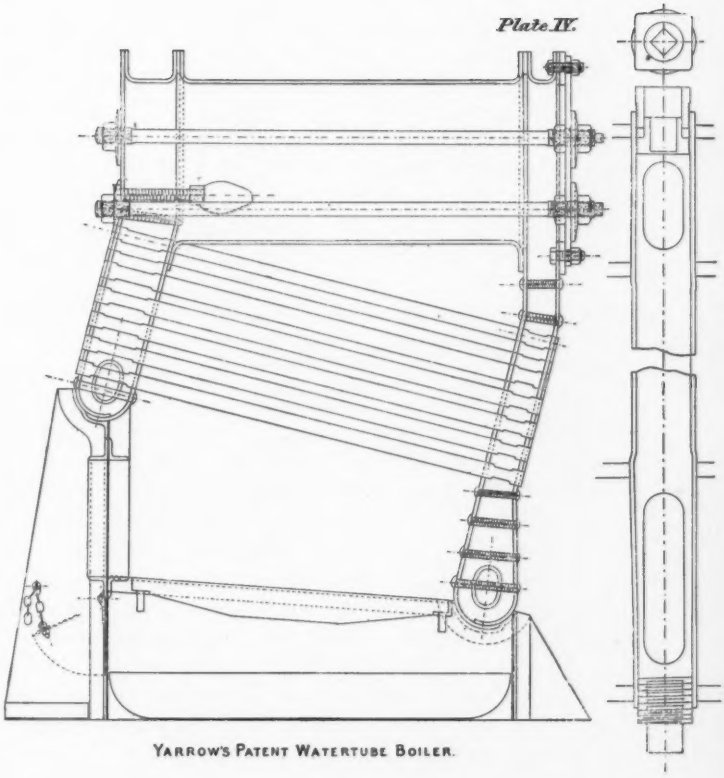
LONGITUDINAL SECTION



HALF SECTION

HALF FRONT ELEVATION
WITH DOOR REMOVED

Plate IV.



YARROW'S PATENT WATERTUBE BOILER.

Besides the boiler described in my previous paper, Mr. Yarrow has used another type, shown on Plate IV. This consists mainly of a horizontal steam and water chamber connected at both ends with water spaces, which are in turn connected by a number of water tubes. In this respect the boiler somewhat resembles the Lagrafel boiler. The construction of the water chambers, however, is different. Each tube passes through both chambers, being expanded into both front and back plates of both spaces. The top and bottom portions of the tubes are cut away at the parts passing through the chambers, leaving the sides intact. The sides, therefore, serve as stays, while the openings left by cutting away portions permit of the circulation of the water. The ends of the tubes are then stopped up, one end being fitted with a screwed plug, the other end being fitted with a gland and packing.

A boiler proposed by Mr. A. E. Seaton is shown on Plate V. It consists of four segments, each connected to a horizontal cylindrical steam chest. Each segment is formed of a horizontal or slightly inclined cylindrical steam and water drum, connected at the front end directly to a deep water chamber. At the back end there is a somewhat similar chamber connected to the drum by means of a circulating pipe which enters the chamber at the bottom. The two water chambers are connected by a large number of slightly inclined water tubes of small diameter. The grate is placed beneath these nests of tubes, considerable space being allowed between the fire and the bottom rows of tubes. The circulation of the gases amongst the tubes is provided for by baffle plates, and by some of the tubes being arranged to touch one another, forming diaphragms. A feature peculiar to this boiler is that no stays are fitted to the water chambers, the outer ends of which are formed of ribbed or corrugated doors. One joint at each end, therefore, exposes all the tube ends similarly to the doors of a surface condenser. This is a point of much importance when considering the accessibility for cleaning or inspection.

Another boiler adopted in the French Navy, and fitted in the *Friant*, *Charles Martel*, *Elan*, and other vessels, and also used in

several steam yachts, is shown in Plate VI. This boiler, designed by Mons. Niclausse, is made by the Cie. Anon. des Générateurs Inexplosibles, of Paris.

The boiler consists of a series of headers fitted side by side, each having a number of compound tubes fitted to it, the whole being placed above the fire and surrounded by a suitable casing. The headers each communicate at their upper open ends with the bottom of a horizontal cylindrical chest, which, when in work, contains steam and water, the water level being at about its center. The headers are made of malleable cast iron, and are each constructed with a center diaphragm dividing it into two portions, the inner serving as an upcast for the mixture of steam and water issuing from the generating tubes, the outer forming a space for the descending water.

For each tube, holes are bored through the front and back of the header, and through the diaphragm of nearly equal diameter, those in the two outer walls of the header being slightly conical or taper, the smaller end of the hole in the outer wall being exactly the same diameter as the larger end of the hole in the inner wall.

The steam-generating tubes are reduced in size at the rear end, and are closed by iron cap nuts screwed on to them, the nuts being slightly smaller than the diameter of the tubes. The front ends of the tubes are secured to malleable iron castings, termed by the makers "lanternes." These "lanternes" are turned with conical surfaces where they fit the walls of the headers. The parts near the tubes which fit the back wall are thick and rigid; the front ends, however, are made thinner, to give more elasticity. The "lanternes" are all turned to gauge, so as to be absolutely interchangeable. The middle part of the "lanterne" fits easily into the hole in the diaphragm; its shell is cut away at top and bottom, so as to afford freedom for the motion of the water in circulating. Inside the steam-generating tube is placed a smaller water-circulating tube, which is secured to a smaller "lanterne" fitted inside the other, but extending only from the front of the header to the diaphragm. The front joint of this inner "lanterne" is screwed.

The tubes are arranged in pairs, each pair being kept in place by a cross girder fastened by a stud screwed into the header. The construction of these details is shown by the sketch in Plate VI, drawn to a larger scale.

In practice it is found that the tubes are readily removable from the front of the boiler, and may be replaced quickly, the boiler thus affording exceptional facilities for cleaning and inspection.

Some tests have recently been made with this boiler under forced draft with very satisfactory results, the steam being produced freely and being found to be dry. Other experiments have been made by feeding the boiler with impure water, but its ready accessibility for cleaning has enabled it to withstand exceptionally bad treatment in this respect without derangement.

A boiler designed by Mr. A. Blechynden, and being made by the Naval Construction and Armaments Company at Barrow, is shown on Plate VII. In general design it presents some features in common with the Yarrow boiler, but it will be observed that the tubes are not straight, all being slightly curved, while the outer rows are made to shield the casing similarly to the outer rows in the Thornycroft and some other boilers. The outside circulating tubes are very small.

A noticeable feature in the boiler is the plan by means of which any single tube may be cut out and replaced from inside the steam chest without disturbing the neighboring tubes.

Plate VIII shows a boiler which has been made and successfully used by Mr. J. Samuel White, of East Cowes. It consists primarily of two lower water chambers and an upper steam chest connected by numerous water tubes, the whole being enclosed in a casing. The grate is below the level of the lower water chambers, a considerable space being allowed between the fire and the lower parts of the tubes to serve as a combustion chamber. The front and back of the casing are shielded by $1\frac{1}{2}$-inch tubes, which are reduced at the ends, so as not to unduly weaken the chambers where they are joined to them, and two longitudinal diaphragms of nearly vertical tubes, touching each other and with similar reduced ends, extend from the front nearly to the back of the boiler, forming return flues. The products of

combustion have thus to pass from the grate down the center of the boiler and return along the side flues to the chimneys. The flues are filled with a number of double spirals of smaller tubes. These are clearly indicated on the engraving. The diaphragm of tubes mentioned as forming the division between the flues is supplemented by baffle plates where required. Very ample external circulating pipes are provided.

The provision for the circulation of the furnace gases by means of the return flues is such as to insure their being brought intimately into contact with the heating surfaces, and to effectually prevent their taking a short cut to the chimney before parting with their heat.

Turning now from the descriptions of the individual types of boilers, let us consider what are the special qualities in which water tube boilers are supposed to surpass the ordinary boiler. It must be remembered that unless the balance of the advantages and disadvantages in any particular case is decidedly in their favor they are not likely to be used, as the preference would otherwise be given to the old, well-tried form. The advantages are mainly:

(1) The means of obtaining higher working pressures than can be obtained with ordinary boilers, owing to the excessive thickness of plates which would be necessary both for the shells and for the heating surfaces.

(2) Economy of maintenance.

(3) Decrease of weight and space required for producing a given power, or an increase of power obtainable with a given weight or in a given space.

(4) Less liability to serious accident through damage by neglect, and also, in case of such damage, less serious consequences, owing to the small quantity of water they contain.

Regarding the first of these considerations, it may at once be conceded that with *all* the types of water-tube boilers very high pressures may be obtained. It is probable that the limit in this respect will, for many years to come, depend not upon the boilers, but upon the engines which have to use the steam after it is made.

The objects of desiring to have steam of high pressure appear to be twofold. First, in order to obtain higher powers with a given size or weight of engine; secondly, as a means towards obtaining greater economy of coal consumption, the greater pressure rendering a greater range of expansion possible, and therefore making the engine more economical. In order to do this, however, it will be essential that the boiler supplying the higher pressure steam should be quite as economical as regards evaporation per lb. of coal as the present boiler, otherwise we shall be losing at the boiler what we gain at the engines. The boiler must also produce dry steam; that is, there must not be a large percentage of moisture in the steam as delivered.

For efficiency of evaporation two things are needed, viz, completeness of combustion, and proper circulation of the products of combustion amongst a sufficient amount of heating surface to enable the latter to abstract the whole of the available heat from the former, which should reach the chimney comparatively cool. These conditions involve both space and weight, and are thus to some extent in antagonism with the requirement of large power on small weight and space. Different types of boilers, therefore, according as the requirements of economy or of large power predominate in the conditions they have to fulfill, provide for these factors more or less completely.

As regards perfection of combustion, it is recognized that so long as coals are used as fuel, sufficient air must be supplied, either through or above the grates, to properly burn the distilled gases from the coal, as well as the carbonaceous part of the fuel; and that the mixture of gases and air require both time and space in which to become thoroughly mixed before their temperature is reduced to a certain critical point, below which further combustion is not possible. In ordinary boilers with roomy combustion chambers these conditions are fairly well met, and in some of the water-tube boilers also these conditions are sought to be obtained; in others, however, in which the products of combustion are hurried direct from the fire into the narrow spaces between the tubes, economy cannot be expected.

Mons. D'Allest, the originator of the D'Allest boiler, attaches

great importance to this point, and his views are borne out by the trials quoted in the Appendix of my previous paper, some of which are reproduced in English figures in Table II. These trials, made by French naval officers with boilers of very similar design in all respects, except that of the circulation of the furnace gases amongst the tubes, and under similar conditions, showed that where the furnace gases rose straight up amongst the tubes, imperfect combustion resulted, smoke of great intensity being produced, while the evaporation was only equivalent to 7.25 lbs. of water, from and at 212 deg., per lb. of coal; in the other boiler, in which the gases passed first into a roomy chamber, where more complete combustion took place, and then traversed the same amount of heating surface, there was no smoke, and an average of 10.76 lbs. of water evaporated from and at 212 deg., a result not only excellent in itself, but remarkable as being almost 50 per cent. in excess of the other. If the natural-draft results only are taken, the average is 11.21 lbs. of water from and

TABLE II.

VARIOUS PARTICULARS OF EXPERIMENTAL TRIALS MADE WITH D'ALLEST BOILERS.

Description of boiler.	Duration of experiment.	Grate surface.	Heating surface.	Total weight of boilers, including all mountings, casings, chimney and water.	Total weight of boilers per square foot of heating surface.	Consumption of Cardiff coal per square foot of grate per hour.	Actual evaporation of water per pound of coal.	Equivalent evaporation from and at 212° F.	Equivalent evaporation per square foot of heating surface per hour.
	Hrs.	*Sq. ft.*	*Sq. ft.*	*Tons.*	*Lbs.*	*Lbs.*	*Lbs.*	*Lbs.*	*Lbs.*
Experimental boiler	6	35.9	1,076			10.29	10.67	12.44	4.28
Experimental boiler	6¼	30.8	1,076			15.15	9.58	11.15	4.84
Experimental boiler	6	35.9	1,076			15.50	9.23	10.83	5.60
Experimental boiler	6	43.0	1,076			15.36	8.97	10.43	6.41
Experimental boiler	3	35.9	1,076			25.09	8.02	9.42	7.86
Experimental boiler	3	35.9	1,076			31.00	8.75	10.28	10.61
Part of installation of vessel of 2,000 I.H.P.	3	Total in vessel.				45.75	8.24	9.39	10.59
Part of installation of vessel of 2,000 I.H.P.	3	100.75	4,090	42.1	23	35.84	9.07	10.34	...
Part of installation of vessel of 2,000 I.H.P.	8					15.36	9.43	10.75	...
Part of installation of vessel of 9,000 I.H.P.	6	Total in vessel.				20.48	9.80	11.16	...
Part of installation of vessel of 9,000 I.H.P.	12	645.	21,528	288.8	30 5	12.30	10.07	11.53	...
Part of installation of vessel of 9,000 I.H.P.	3					24.60	9.50	10.86	...
Part of installation of vessel of 9,000 I.H.P.	3					30.72	9.23	10.58	9.70

Boiler proposed by Mr. A. E. Seaton.

PLATE V.

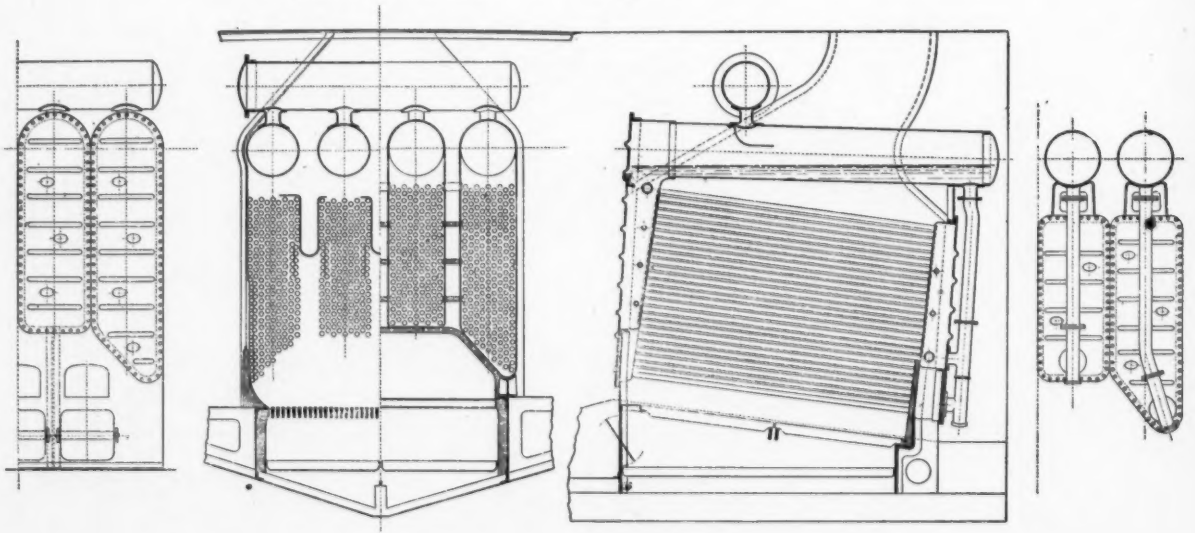
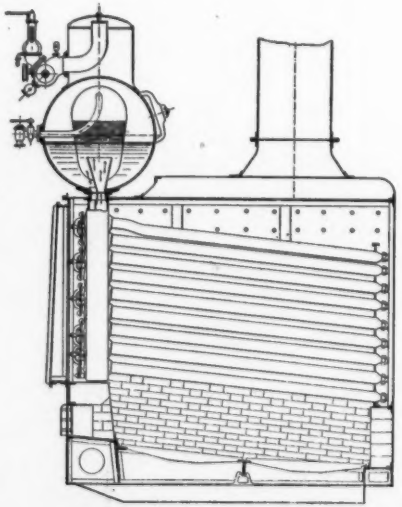
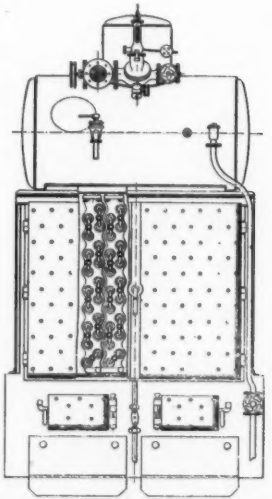


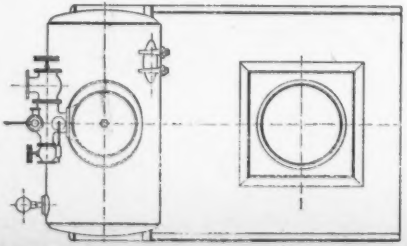
PLATE VI.

NICLAUSSE BOILER

MADE BY SOCIETÉ ANONYME DES GENERATEURS INEXPLOSIBLES, PARIS.

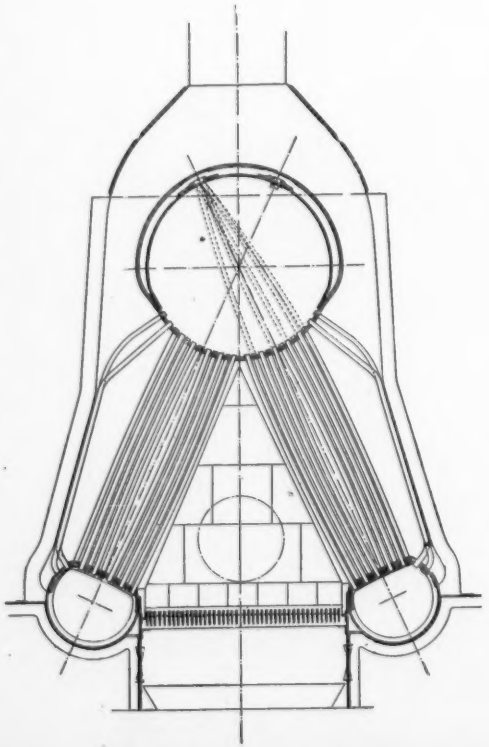


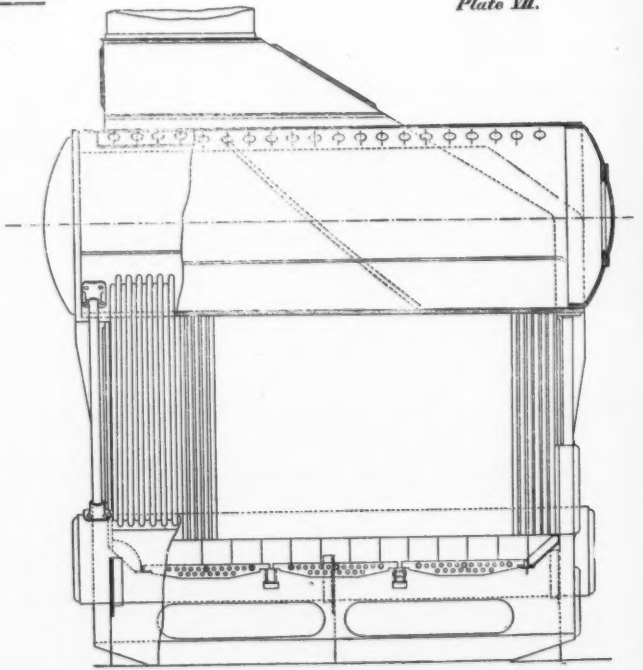




BLECHYNDEN'S
WATER-TUBE- BOILER.

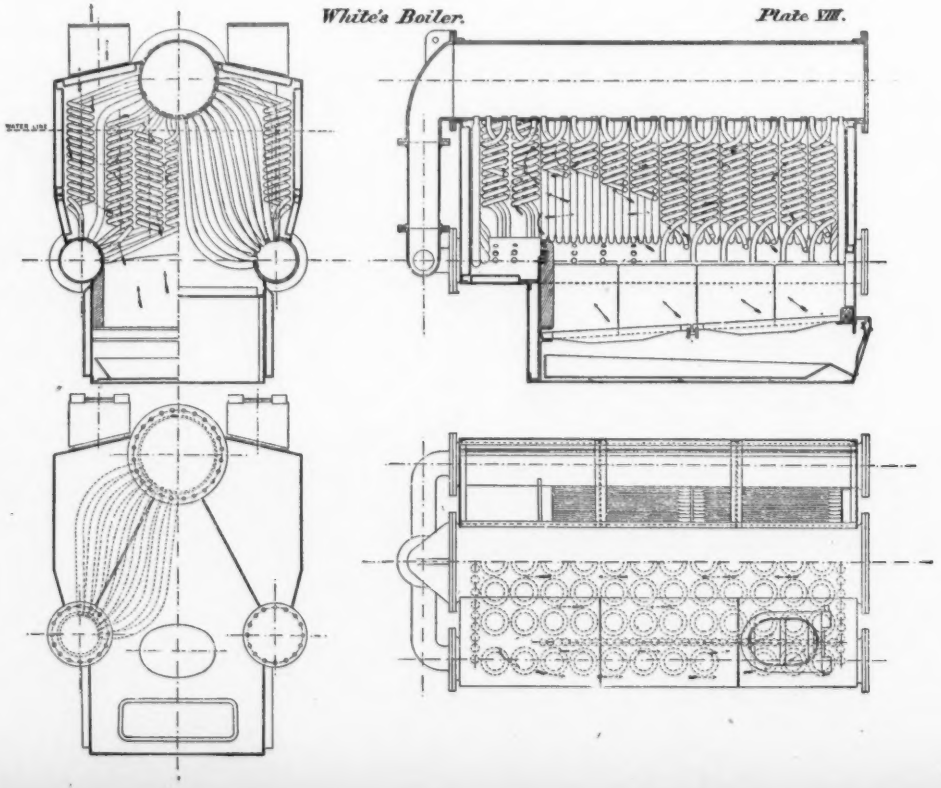
Plate VII.





Scale 1½"=1 Foot

White's Boiler. *Plate VII.*



at 212 deg. Attention is also called to the results obtained with forced draft in the cases of the 2,000 and 9,000 I.H.P. vessels. These experiments are very valuable, being made by French naval officers, entirely independent of the makers of the boilers, and therefore disinterested. The importance of providing for proper combustion and circulation of the furnace gases amongst the heating surfaces is shown also by the results of the experiment made by Mr. Stromeyer on Messrs. Anderson and Lyall's boiler, the results of which have already been given.

Under the head of economy of maintenance, many points have to be considered, some bearing generally upon design, others on details, and the question also of the size and material of the tubes is important.

Regarding the material of the tubes, it appears that, at least with the smaller tubes used with the very light type of boiler, seamless steel is the most suitable, although seamless copper and lap-welded iron tubes have been used. The seamless tubes are preferred, not because of the relative weakness of the weld in lap-welded tubes—for even welded tubes, of the small sizes used, possess an enormous margin of strength in proportion to the working pressure—but in order to obviate the possibility of minute imperfections in the weld, which have the effect of reducing the thickness available for resisting corrosion, it having been found that occasionally an almost imperceptible corrosion would open a minute hole through the tube at such a defect, completely spoiling it. Undoubtedly seamless tubes would possess the same superiority over welded tubes, even for the larger sizes of water tubes.

While recollecting the good work which iron lap-welded tubes have done in the past, as ordinary boiler tubes, it should be remembered that the corrosive influences to which they are subjected, are on the outside, where the weld is more likely to be perfect than the inside, and where, if any imperfections exist, they are at once visible. If water-tube boilers come into general use, however, our tube-makers must supply us with seamless

tubes at a reasonable price. With the demand for them, no doubt they will be forthcoming.

Next, with the small tubes, comparatively thin tubes must be used; but with larger tubes this is not so necessary, and the greater thickness will give larger margins against wear and tear and corrosion, than can be obtained with smaller tubes. In most of the types of boilers, the most vulnerable parts appear to be the tubes, if we except, perhaps, the furnace fittings, which are in general not dissimilar to those of other boilers. In most of the boilers it is a comparatively simple operation to renew any individual tube, although in some boilers it would appear to be necessary to remove several tubes to obtain access to others.

Another point, as to which great differences present themselves amongst the various types, is the facility for examination and cleaning. In this respect those boilers have the advantage in which the tubes are straight, and of sufficient size to enable them to be looked through. Those also in which an examination can be made without the breaking and remaking of numerous joints have an advantage over those requiring one or two joints to be broken for every tube that has to be examined. Then, again there are joints and joints; some are metal to metal joints, others require jointing material. The former would in general appear to require more skill and care in making than the latter, but would probably be more satisfactory and permanent when once properly made.

The above remarks as to joints apply only to what may be called door joints. The joints between the tubes and water chambers, or between different lengths of tubes, again vary in the different boilers. Some engineers prefer ordinary rolled or expanded joints; others demand screwed joints. In some of the boilers, with zig-zag arrangement of tubes, the number of screwed joints at the bends or elbows is very great.

Some of the boilers described may be made equally well of large or of small size. For instance, the Belleville boiler may be made with as few as four elements, or a much greater number may be employed. In small vessels, where it is required to sub-

divide the power, it will be advantageous to make them of small size; but in large-powered vessels, and in those in which it is desired always to work at about the full power, great subdivision of boilers appears to be a mistake, a less number of larger boilers being more easily attended, and requiring less complication of fittings and pipes.

An important point bearing upon the question of durability is the possibility of keeping the outsides of the tubes free from soot and from the accumulation of fine ashes, the latter, especially when moisture is present, being very destructive to iron structures. Great differences exist amongst the various boilers in this respect. Another point which must not be overlooked is whether the design provides for a depositing chamber in which the impurities of the feed water will accumulate. Although it is fully recognized that these boilers demand absolutely pure water for feed, and evaporators and filters, &c., are supplied, yet it is inevitable that some sea water and other impurities will occasionally be introduced into the boilers. These appear, in the main, to become separated out from the water into a solid form more at the time the water is being raised in temperature to the boiling point, than during the time the water is being evaporated. Those boilers would appear to have an advantage which provide, in the course of the circulation of the water, quiet places in which the impurities can settle without encrusting the heating surfaces.

The importance of the proportions of the weight and space occupied to the power of the boilers varies much in different vessels. In ordinary cargo steamers both considerations are comparatively unimportant. The first consideration in these vessels is economy of coal consumption, and the boilers which give the best results in this direction will be preferred to others, even if they occupy more space and are considerably heavier. In passenger steamers, however, the weight and, more especially, the space occupied are important; but in these vessels also, especially those engaged in long voyages, economy of coal consumption is still of vital importance.

In war-vessels, however, the case is very different. What may be termed the weight efficiency and space efficiency assume the first importance, the questions of economy and durability taking relatively second place. In some cases, indeed, the question of economy hardly comes in at all, the question being mainly that of obtaining the greatest power on the smallest possible weight and in the small space available. In other cases, however, combined with the question of large power is the requirement of economy of consumption at low power. It is evident, therefore, that the same type of boiler can hardly prove to be the best for all purposes.

ON THE COMPARATIVE MERITS OF CYLINDRICAL AND WATER-TUBE BOILERS FOR OCEAN STEAMSHIPS.

By James Howden, Esq.

[Abstract of paper read at the Thirty-fifth Session of the Institution of Naval Architects, March 15, 1894.]

A comparison of these distinctive types of boilers appears to be not inappropriate at the present time, which witnesses another revival of the water-tube boiler for sea-going steamers, after the frequent, but hitherto abortive attempts, which have been made for this purpose during the last thirty-five years. This revival of the water-tube boiler is the more remarkable because of the claims made by its supporters of its superiority to the cylindrical boiler, whose speedy extinction by it some, at least, confidently predict. This hope or expectation, for it is as yet unrealized, of the new school of boilerists, is also remarkable in being entertained in view of the following facts regarding the cylindrical boiler:

(1) In being entertained, after thirty years of uniform, unrivalled and continuing success of the cylindrical or "Scotch" boiler, for every usable or practicable steam pressure yet required in sea-going steamers.

(2) In being adopted at the period when the cylindrical boiler is still rising in efficiency, and when examples of single boilers of the type, in moderate dimensions, supplying steam of 200 pounds working pressure for 2,500 I.H.P., are being worked at sea without the slightest feeling of risk, damage or inconvenience.

(3) In being advocated in view of the indubitable fact that the design and introduction of the cylindrical boiler was chiefly owing to the failure of the water-tube, sectional, and all other forms and types of boilers to supply sea-going steamers with high-pressure steam safely, easily and economically; and in the

face of the still more important fact that, from the day of its introduction, all the serious troubles and vexations previously caused by the other types of boilers named came to an end, and the cylindrical boiler quickly became the universal type for steamships solely by reason of its unrivalled merits.

[Mr. Howden then goes at length into the early history of water-tube boilers, the increase in boiler pressure, and the successful introduction of the cylindrical or Scotch boiler.]

(1) So strong is the shell of a cylindrical boiler built of steel plates and riveted in the usual manner, that even if made of half the thickness required by the Board of Trade it would be impossible to burst it, however large in diameter, by any steam pressure that could be raised within it. The percentage of elongation required to break boiler steel is so great that before one-fourth of it could be reached by undue internal pressure each rivet and joint would become a safety valve and effectually prevent bursting.

The furnaces are likewise of the most perfect form to resist collapse under pressure. The several well-known forms of furnaces used in marine boilers for the highest pressures, and for diameters over 4 feet, prove that even the largest size can be safely used. Should it be said that the furnaces sometimes collapse, and that this is a weak point in the cylindrical boiler, I admit that furnaces do sometimes collapse, owing to the plates above the fires overheating from deposit of grease or heavy incrustation, but I have never heard of one such case where any rupture of the furnace plating has occurred so as to create danger. The more careful treatment of the feed water now common should make these cases more and more rare, but absolute security against collapse can be obtained by the use of ringed furnaces, where the pitch of the rings is reduced somewhat as the pressure increases. Even with very great overheating of the furnace plates, only slight bulging can be produced in these furnaces, and collapse is impossible with water in the boiler. The tubes of the cylindrical boiler, even when a leakage occurs, cause little trouble and no danger. Plugs can be pushed in from the smoke-box end to stop most of the leakage, if not

all, and it is generally not necessary to do more until arriving at a port, when, if a spare tube is not at hand, a tube stopper will enable the boiler to run for months without replacing the defective tube.

(2) It contains within one enveloping shell its water, steam, fires and fire gases, while the heat generated, until it leaves the boiler, is entirely utilized for its proper purpose of evaporation by being surrounded by water from first to last. This feature ensures a more effective heating surface and greater proportional economy than can be attained by any other type, such as the water-tube boiler, not possessing this feature.

(3) The fire gases act on the smaller interior surface, while the water receives the heat on the larger exterior surface of the communicating plating. This feature ensures a quicker and safer evaporation than when the conditions are reversed, as in the water-tube boiler.

The effect of these two latter distinctive features of the cylindrical boiler and boilers of similar class is, that from the combustion of an equal quantity of coal, and with an equal heating surface, a considerably greater evaporation, or power, will be obtained from the cylindrical than from the water-tube boiler, while a much higher rate of combustion can be safely utilized in the former than in the latter type.

The greater economy of the cylindrical boiler here claimed is proved by the fact that an equal, if not greater, evaporation per pound of coal is obtained in this boiler from two-thirds to one-half the amount of heating surface than that required in water-tube boilers. This will be clearly shown from examples given further on. In proof of a much higher rate of combustion being more safely utilized in the cylindrical boiler, I point to the fact that from the very small surface of the furnaces of a cylindrical boiler, if of fair length, about one-half of the total evaporation is obtained with absolute safety, in some boilers even for twenty years, without injury, from the same furnaces. I may point to the analogous case of the locomotive furnaces, though not nearly so favorably disposed as the semi-circular form of the cylindrical boiler, in support of the astonishing extent of evaporation per

square foot of surface which can be safely taken from such furnaces. The limit of safe quantity in the cylindrical boiler has not yet been half tested.

Examining now the water-tube boiler and its distinctive features, we find that it is composed of a large number of separate parts having a separate connection with a common feed-water vessel or vessels at their lower ends, and with a common steam receiver or receivers at their upper ends. Further, that the heat is received on the outside or larger plate surface, varying more or less according to the diameter and thickness of the tubes. The evaporating action is, therefore, distinctly less favorable in this type of boiler, being in radial lines towards the center of the tubes, and in a confined space where the lines of motion converge on and repel each other, while the passage area for escape of the globules of steam being very limited, the tube surface, exposed to a high heat, cannot be so well protected by the water from its action as it is in the cylindrical boiler, where the quantity of water is large and common to the whole boiler, and where the globules of steam evaporated have vastly freer means of escape. That these features in the water-tube boiler make them less effective per square foot of heating surface, and less durable than the other, is abundantly proved by experience.

If any water-tube boiler is worked at a high rate of combustion, such as is quite safe in a cylindrical boiler, say to give an evaporation of 10 pounds of water per square foot of heating surface, two results will inevitably follow: (1) If it is found capable of reaching this evaporative power, and I find no case of its being nearly approached by this type of boiler, it must have a short life and be utterly unfitted for any sea-going steamer, and (2) it must consume an enormously greater quantity of fuel than the cylindrical boiler per pound of water evaporated. It so happens that in the water-tube boiler, even a moderate evaporative power, combined with a moderate economy, can only be had under conditions which shorten the life of the boiler, and prolonged existence, with a moderate economy, can only be secured by a greatly reduced evaporation per square foot of surface, and consequently a large increase in size and weight of boiler. As these

alternatives proceed from essential features pertaining to water-tube boilers when working, I may be pardoned for occupying time in further explanation of these features and their consequent results.

The small tube-boiler, while much more efficient than the large tube boiler for mere purposes of evaporation, is, however, quite unfitted for ocean steamers. The chances of damage are in proportion to the number of its tubes. The thickness of the tubes must necessarily be limited in small diameters. The high temperature acting on the tubes next the furnace, which from the small sectional area of the contained water and its rapid evaporation, must always be filled with steam and water combined, and working under conditions fatal to longevity, while these, and the other tubes, are working under the continual risk of foulness of water, whether from grease, salt or other impurities, which are present in all marine boilers more or less, whatever precautions are taken, and which, sooner or later, will act upon the material of the tubes. If they are thin, they must go quickly, if thick, with proportional increase of water area, they will endure longer; and, if cap able of being well cleaned out at short intervals, their durability will be still further increased.

Having said so much regarding the distinctive constructive features of these two types of boilers, and the action of the fire-gases in each, I must at this stage separate the class of water-tube boilers for torpedo and other light high speed steamers and the class for ocean steamships, with which latter I have chiefly to do. The former class, of which those designed and construced by Mr. Thornycroft and Mr. Yarrow in this country, and by Mr. Ward and others in America, give wonderful proof of what can be accomplished for short period, by great ingenuity combined with first-class workmanship and materials; but these boilers are quite unfitted for working at sea. The great number, smallness, and closeness of their tubes, on which their steaming qualities depend, while absolutely necessary to take up, with the necessary rapidity, as I have explained, the high heat from the rapid combustion of a large quantity of fuel, unfit them for continuous work at sea. If one tube gives way, the whole

contents of the boiler will find egress therefrom, and cannot be stopped. The boiler must then cease work, and be practically taken to pieces before the tube can be withdrawn and replaced. These boilers are suited for short periods of working only, and could not continue running at high power for days together at sea. There is a tendency at the present moment, both in this country, on the Continent, and in America, all more or less under the prevailing water-tube epidemic, to overrate these performances. The following table, compiled from actual performances of vessels at sea, by Mr. McFarland of the United States Navy Department, the talented and courteous Secretary of the Marine Engineering and Naval Architecture Section of the Congress at Chicago, shows that even these small water-tube boilers under high forced draft do not put the locomotive type, which is analogous to the cylindrical type in its action, much in the shade in the matter of weight per indicated horse-power, while in all, or mostly all, other points the locomotive boiler is superior.

COMPARISON OF VARIOUS TUBULOUS AND SHELL BOILERS.

Kind of boiler and maker	Tubulous Thornycroft.			Shell, cylindrical double ended.	Shell, locomotive.	
Where used	U. S. S. *Cushing.*			U. S. S. *Newark.*	Italian torpedo cruisers, *Tripoli* and *Folgore.*	
Outside dimensions	10 ft. × 7 ft. × 8 ft.			13 ft. 6 in. dia. 19 ft. 5 in. long	16 ft. 8 in. × 6 ft. 4 in. × 7 ft. 6 in.	
Grate surface, square feet	38			135	28	
Heating surface, square feet	2,451			4,185	1,116	
Ratio heating surface divided by grate surface	64.5			30.99	39.8	
Weight of boiler, empty, tons	9			57.8	.....	
Weight of boiler and water, tons	11			80.12	15.6	
Duration of trial, hours	2.5	11.5	1	8	.....	.....
Air pressure in inches of water	*0	3	4	2.25	3.13	4.95
Coal per hour, per square foot grate surface, lbs.	7.58	40.23	66.32	40	98.3	120.8
Evaporation from and at 212° Fah.	11.9	8.84	6.51	.....	6.97	6 62
Actual evaporation from and at 212° Fah. per square foot heating surface.	1.4	5.51	6.7	Water not measured.	17.2	20.09
Horse-power per 100 square feet of heating surface on basis of 20 lbs. steam per hour from and at 212° Fah.	7	27.55	33.5	55.9	86	100.45
Horse-power per ton of boiler and water on same basis	15.6	61.38	74.65	29.2	61.52	71.85
Horse-power calculated from above particulars	171.6	675.18	788.15	2,339.5	959.7	1,120.86

*Blowers discharging into open fire-room.

With water tube boilers capability of working at sea depends on the tubes being of a diameter large enough to contain a sufficient quantity of water to prevent rapid destruction from the heat, and of sufficient thickness to withstand rapid destruction from the water either in its friction or impurity. Another essential condition in water-tube boilers for ocean steamers is that the boilers be subdivided to such an extent that, in the event of an injury to one section, it could be shut off without greatly reducing the total power. Further, provision should be made for replacing injured tubes without practically having to take the boiler to pieces.

I may be allowed, however, in comparing these boilers of the class capable of working at sea with cylindrical boilers to select the Belleville boiler for this purpose, as, in its latest form for sea-going steamers, it fulfills all the conditions I have specified as necessary for such steamers. The selection of the Belleville boiler for comparison must not, however, be taken as a reflection on the merits of any other form of this class of boiler.

That the conditions I have specified as necessary in water-tube boilers for sea-going steamers have been adopted by the Belleville Company shows that they have found them, from experience, to be necessary. And I would here remark that, in examining critically the design and arrangement of the Belleville boiler, it is impossible to withhold one's admiration of the careful study which has been given to every point of detail, and for the provision made for minimising, as far as possible, the special dangers pertaining to this type of boiler. The tubes in the latest Belleville boilers for ocean steamships are $5\frac{1}{4}$ inches external diameter, and from $\frac{3}{8}$ inch to $\frac{1}{4}$ inch in thickness. The boilers are subdivided into numerous sections, each of which can be detached from those adjacent, and special provision is made for the replacement of injured parts. I understand that there is a special arrangement of this boiler for war ships, with tubes 3 inches in diameter; but as it appears that in all countries alike durability, safety and permanent efficiency are sacrificed in such ships to erroneous ideas of lightness and high trial power, I shall not further refer to this form of the Belleville

boiler. The makers, from their wide experience, have had good reasons for adopting the large size of tubes they now use in boilers for ocean steamships. They are the only boilers I am aware of that have made long ocean voyages.

Examining this boiler in its action and utilization of the fire gases, it will be evident, for the reasons set forth, that these large water tubes cannot be worked at a high rate of combustion without a great waste of fuel arising from the size and disposition of their tubes and their reduced efficiency per unit of heating surface. Even at what would be a low rate of evaporation per unit of heating surface in a cylindrical boiler, this water-tube boiler would allow a large portion of heat to escape up the chimney, and by radiation from the casings, if not specially protected.

Coming to actual performances, I find that the steamers *Polynésien* and *Armand Behic* are reported to have attained on trial 8,000 I.H.P., while their average working power at sea is 5,000 I.H.P. The respective grate and heating surfaces in these boilers are 580 and 23,800 square feet. We have thus the following results:

	Trial.
Indicated H.P. per square foot of grate	13.62
Heating surface per I.H.P., square feet	2.975
	Sea.
Indicated H.P. per square foot of grate	8.62
Heating surface per I.H.P., square feet	4.76

I have no information as to fuel consumed, but, as the draft is directly upwards from the fire to the chimney, it must necessarily be high, and will require to be greatly checked to prevent waste of fuel. The consumption may probably be from fifteen to sixteen pounds per square foot of grate at sea.

The boilers as fitted in these steamers are twenty in number, placed back to back and side by side, and fired from the wings. There are twenty furnace doors in each stokehold, or forty in all, but each two doors give access to a common furnace grate, so that each boiler has 29 square feet grate surface and 1,190 square feet H.S.

The space occupied by the boilers lengthways in the ship is 70 feet, and 14 feet athwartships, with a height of 13 feet 4 inches,

not including uptakes. The weight, including water, all mountings and funnel, is 380 tons.

These particulars enable us to compare their power, weight and the space they occupy in the ship with (1) the size, power and weight of cylindrical boilers capable of working continuously at the highest power attained by the Belleville boilers on trial—that is, 8,000 I.H.P.; (2) the size and weight of cylindrical boilers capable of working with ease and economy at the sea power of the Belleville boilers—that is, 5,000 I.H.P., but which can be worked as high as 6,000 I.H.P. when required. The cylindrical boilers I assume as working with my system of forced draft, it being easier, more comfortable, more economical and involving less wear and tear on the boilers than natural draft.

These cylindrical boilers capable of maintaing 8,000 I.H.P. are three in number, double-ended, 14 feet 6 inches diameter by 19 feet 6 inches in length, having six furnaces each 3 feet 7½ inches inside, or mean diameter, or eighteen furnaces in all. Aggregate grate area, 360.5 square feet, and heating surface, 14,000 square feet; 8,000 I.H.P. from these boilers gives 22.22 I.H.P. per square foot of fire grate, and 1 I.H.P. from 1.75 square feet of heating surface.

I only mention here the fact that from furnaces of the same size there has been maintained over several years, in ordinary course of working at sea, equal rates of power per square foot of fire grate and per square foot of heating surface.

The space occupied by these boilers with their non-conducting covering is 45 feet by 19 feet 6 inches, or 877.5 square feet floor space, against 980 square feet floor space required for the Belleville boiler, or 102.5 square feet less space for the higher powered cylindrical boilers. The space occupied by the cylindrical boilers is actually still less in comparison when the stokeholds are taken into account. Those required for the Belleville boilers are two, 70 feet in length, and, as space must be given across the stokeholds to allow the parts of the boilers to be removed and repaired, the stokeholds must be quite equal in width to those required for the cylindrical boilers. Taking the width in both cases at 8 feet, and their lengths equal to that occupied

by the respective boilers, we have for the Belleville boilers $(14 + 8 + 8) \times 70 = 2{,}100$ square feet of occupied boiler-room area, and for the cylindrical $(19.5 + 8 + 8) \times 45 = 1{,}597.5$ square feet, or only .76 of the space required for the Belleville boilers. These cylindrical boilers are, however, far beyond the power of the water-tube boilers. Taking their normal power at sea, as stated by Mr. Milton in his paper read at Cardiff, last year, at 5,000 I.H.P., then two cylindrical boilers, 15 feet 6 inches diameter by 20 feet in length, will be amply sufficient for this power, each boiler having six furnaces 3 feet 9 inches inside, or mean diameter; grate surface, 250 square feet; heating surface 10,000 square feet. The indicated horse power per square foot of grate in this case is only 20, and the heating surface as high as 2 square feet per indicated horse-power, leaving a considerable reserve of power available when required. The space occupied by the boilers in this case is 32 feet by 20 feet, or 640 square feet, as against 980 square feet, and, if the same width of stokeholds be taken for these boilers, we have $(20 + 8 + 8) \times 32 = 1{,}152$ square feet, against 2,100 square feet for the Belleville boilers of same power. Instead of therefore occupying less space in the ship, the Belleville water-tube boilers, in comparison with the cylindrical of same power, occupy space as 1.82 is to 1.

In calculating the weight of the cylindrical boilers, I have taken their maximum pressure at 175 pounds, to ensure a constant working pressure at 170 pounds, Board of Trade rules being followed.

The weight of the three boilers to maintain at sea 8,000 I.H.P., including the fans and engines and all forced-draft apparatus, water, funnel, and all mountings, as in the water-tube installation, is 361 tons, or 19 tons less than the lower powered Belleville boilers.

The total weight of the two cylindrical boilers with water, all mountings and forced-draft fittings complete, capable of maintaining easily the same power at sea as the Belleville boilers, economically and safely, is only 272 tons, or 108 tons less weight than the Belleville boilers. When examined, therefore, on the

points in which this boiler is supposed to excel, space occupied, and weight, it comes very far behind the cylindrical boiler when worked in the most effective manner.

There is, however, in connection with this comparison of weights, the very important item of the weight of the water in the boilers of these two types. Though I have no means of calculating the weight of the water in the Belleville boilers, it is probable that it is not more than one-third that of the cylindrical boilers, and about five-twelfths that of the two cylindrical boilers. But for the much smaller quantity of water carried in these tubulous boilers they would compare still worse in regard to weight than they do, for the necessary brick work and the enclosing iron casings, if got up in a durable manner, and so as to effectively prevent escape of heat from the furnaces and casings in boilers for sea-going steamers, is often heavier than the shell plating of a cylindrical boiler of equal power. It is often overlooked that the specific weight of fire-brick is more than one-fourth that of iron, so that a 9-inch wall of fire-brick is as heavy as $1\frac{3}{8}$ inch steel plate per square foot of surface. It is not, however, sufficient to have only a brick wall to prevent escape of heat. This must again be covered by a considerable depth of non-conducting substance enclosed by an iron casing.

In these tubulous boilers the aggregate weight is thus so far saved by the reduced weight of water carried. This is claimed by its advocates as a great advantage, but to practical engineers concerned in working steamships across the ocean with ease, safety and speed, this feature will be regarded as a serious defect. The cylindrical boiler owes no small part of its durability as well as its safe, steady and effective working, to the large proportion of water to heating surface it contains.

The practice of the makers of the Belleville boiler, who have undoubtedly given most careful attention to all the weak points which their experience has shown attend the use of these boilers, is quite against the conditions Mr. Seaton desiderates, as they appear to use not one-tenth, but about one-third or so of the water in their boilers for mercantile steamers that would be used in a cylindrical boiler of same power. Yet they have evi-

dently experienced difficulty in the regulation of the feed water. I am informed that they now use an automatic regulation of the feed water, which requires only one attendant in such steamers as the *Armand Behic*, with her twenty separate boilers. If this is so, and this attendant is not assisted by others in the stokeholds, this automatic feed regulator is a marvel indeed, and the attendants must be specialists of a very high order. What may be possible, however, as regards regulation of feed in these French steamers, maintaining, as nearly as possible, uniform speed and power from day to day, with trained specialists in charge, would be impossible and dangerous with ordinary men picked up promiscuously, as is almost universally done among the numberless steamships of this country. If one-twentieth of the steamships now running with cylindrical boilers were fitted with Belleville boilers, the casualties from this one difficulty with the feed water would, I am convinced, be legion.

Taking the case of war ships and cruisers when in actual warfare, with speeds varying rapidly from fullest possible to dead slow, with most variable stoppages and startings, automatic feed regulation would be utterly deceptive, and dangerous in the highest degree, without a man being in exclusive attendance on every two or three boilers at most, and, if a difficulty did once occur, it could not fail to multiply quickly. What this means on a large vessel, with forty or fifty separate boilers on board, can be easily imagined. A degree of safety attainable with these boilers in a merchant steamer with attention would be impossible of accomplishment in a war ship at the very time when efficiency was most needed. I have, therefore, been unable to come to any conclusion as regards war ships than that even these tubulous boilers with large tubes are quite unsafe as well as unsuitable. They may run trials with much success, and likewise may cruise at a steady speed for a lengthened period without mishap, but for meeting the conditions of actual warfare, they are, in my opinion, quite unsuited.

It has been also claimed that the quick steam-raising properties of the water-tube boilers would be a great advantage in a war ship when cruising in time of war and suddenly called upon

to steam at full power. The supposition, I believe, in this case, is that the war ship would have only some of her boilers under steam when the emergency arose, the greater part having merely the fire bars coaled ready for lighting. I can well understand how difficult it would be to maintain fires in water-tube boilers, even in the lowest possible state of combustion, when only a fourth or fifth of full power supply was being required. The difficulty with water-tube boilers would then, doubtless, be so great that only those boilers required for the actual steam supply would be used, the others being merely prepared for lighting. This would, however, be a most dangerous condition for a war ship to be in when, from sudden attack or other contingency, command of full power was of vital moment. What advantage would it be, although full steam could be got up in such water-tube boilers in an hour or an hour and a half after lighting fires? Ten or fifteen minutes, or less, would often be enough to decide loss or safety, victory or defeat, vital manœuvers carried out or frustrated.

As regards safety, I may point out that, in water-tube boilers, if one tube is injured, or any of the numberless joints give way, the boiler is entirely disabled, as all the water and steam in the boiler will find egress from the rupture if the fires are not drawn, which often may be impossible to do. The bursting of a single tube is a much more serious thing in a water-tube boiler than in a cylindrical boiler, and has every chance of being exceedingly dangerous, for under the high pressures now used the outflow may be so rapid as to fill the stokehold with steam, and necessitate the immediate exit of the attendants, if that is possible.

Further, as is well known, every fitting, such as feed check valves, water gauges, with their cocks and pipes, stop valves, safety valves, blow-off cocks, and the numerous steam and water pipes connected therewith, creates an additional risk, and requires also additional care and attention. How much more danger, therefore, actually exists, and how much more care and attention should be necessary in the case of the *Polynesien*, and the other French steamers, in which there are twenty boilers and twenty sets of each of the different parts I have enumerated,

than exists or would be required for the two or the three cylindrical boilers equal in power to the twenty boilers, in which latter two sets of each only are required in the one and three sets in the other?

The fact of the present "boom" in water-tube boilers for Naval purposes does not alter the facts regarding them which I have presented, and which will ever, I believe, militate against their adoption for mercantile steamers, though the admirable arrangement of detail, precautions in regard to feed water and facilities for repairs, together with the limited size and power of each boiler, and the considerable thickness of tubes, which the makers of the Belleville boilers have adopted for ocean mercantile steamers, will doubtless secure for them a considerable lease of life, and minimize some of the disadvantages I have pointed out. Yet these disadvantages being inherent in this type of boiler, they can never successfully compete with the cylindrical boiler.

The country in which water-tube boilers are most largely used is the United States of America, thousands being at work on land, and a number, not yet large, in the river and lake steamers, chiefly of the steam launch size. It can be easily understood that in that great country, where the transport of large heavy articles inland is difficult and most expensive, if not impossible, there is a most legitimate field for the use of water-tube and other sectional boilers, consequently they are very largely in use there, and have been for many years. Notwithstanding this, I have never found any American so venturesome as to put such a boiler on board a sea-going steamer. Shipbuilders and engineers on Eastern seaboard, like Messrs. Cramp & Sons, or the Union Iron Works of San Francisco on the Western seaboard, who build and engine sea-going steamers, have never used water-tube boilers for such steamers. If the great advantages of this type over the cylindrical boiler which its advocates in this country affirm belong to it were believed in by these practical Americans, you may be assured they would have been filling their ships with them by this time. No people are less bound by custom or tradition than Americans, or readier to adopt anything they believe to be superior to what they have

been previously using. As they have had for many years plenty examples of water-tube boilers of most ingenious and workable designs, the absence of these boilers in any American mercantile sea-going steamer is significant.

I am aware that one of the steamers of the United States Navy, the *Monterey*, has fully two-thirds of her steam power in four water-tube boilers; but I am not aware that they have as yet been put to any trial extending over many hours of work at one time. The Engineering Bureau of the Naval Department have wisely fitted in that steamer, in addition to the coil boiler, two "Scotch" boilers. These two boilers have as yet, I believe, done all the cruising. The Americans mix a good deal of common sense with their boldness. The water-tube boilers used in the *Monterey* are Ward's coil boilers, a very ingenious design, especially the feature where the steam generated in each semi-circle of the coil is relieved by vertical tubes of larger section, through which it obtains access to the steam receivers above. The boiler is, undoubtedly, among water-tube designs a first-class generator of steam, but unquestionably its proper field of action is on shore, and not on board a sea-going steamship.

I have thus endeavored to set forth the comparative merits of cylindrical and water-tube boilers for ocean steamships; and, I believe, that all who will examine this subject without prejudice must come to the conclusion that the cylindrical is by far the most suitable.

Discussion.

The discussion on the two preceding papers was combined, and from abstracts, as well as subsequent letters on the same subject, published in "Engineering," the following opinions and statements have been compiled:

Mr. Seaton said his boiler had been at work nine months; on the trial a tube had split, but since then there had been no trouble. He had found it necessary to reduce the fire grate, which had formerly been as large in proportion as those used in the Thornycroft or Yarrow boiler. This reduction in grate area

had led to better economy. When making the alteration some tubes were drawn; on the outside a bituminous deposit was found, as if the surface had been tarred; inside there was a slight deposit of mud, but the tubes were what he would call clean tubes. With regard to curved or straight tubes, he preferred the latter. With curved tubes one could not see from end to end; with straight tubes it was possible to do this. He did not believe in any boiler where it was necessary to destroy a lot of tubes to replace one. Neither did he believe in the "priming boiler"; it did not seem to him good engineering to boil up the steam and the water together, trusting the former to go to the engine and the latter back again. The construction of water-tube boilers was now possible in consequence of the improvement in the manufacture of tubes, which could now be obtained solid drawn.

Mr. Yarrow said that it was about seven years ago that his firm first commenced adopting straight tubes in their water-tube boilers, having been previously led away by what he believed to be the popular fallacy that curved tubes were necessary in order to allow for contraction and expansion. No doubt it was necessary to curve the tubes in some designs, but it did not follow that it was necessary in all.

Although his firm had built a large number of water-tube boilers during the last seven years with straight tubes, it might be of interest to the meeting to know that they had never had a single leakage between the tube and tube plate. This fact proved, they believed, that there was nothing objectionable in the straight-tube system, whilst it clearly offered many advantages, both in construction and in service. Experience had shown that steel tubes, whether galvanized or not, rapidly deteriorated, and if the tubes were galvanized on the inside a rough surface was caused, which, in some instances, retarded the circulation of the water. There was also the possibility of portions of spelter lodging in the tubes and partially blocking up the passage, or causing such an obstruction which would allow sediment to collect. So far as his experience went, it led to the

conclusion that copper tubes were more durable than steel, but in the use of copper it was essential that it should be very pure, and that the tube, under all circumstances, should be below the level of the water and always filled, otherwise the copper might become overheated. Mr. Thornycroft had given the meeting valuable information concerning the circulation in tubes which are drowned, and also in those which lead at their upper extremity to the steam space. In spite of the possible reduced circulation in drowned tubes, referred to by Mr. Thornycroft, the circulation was still ample for the purpose, and any more rapid circulation was unnecessary. He would state that his firm commenced experiments with water-tube boilers as far back as 1877, and put a water-tube boiler in a torpedo boat in the year 1889, and their experience indicated that very small changes in design materially affect the action of a boiler; small alterations in some cases making the difference between success and failure. If portions of the boiler be rigidly secured by stays, which are not subject to the same temperature as the tubes, serious strains may be set up, involving continual bending of the tubes to meet continual variations of temperature. If, in such cases, the tubes be only slightly bent, it might lead to trouble, as small alterations in length involve considerable alterations in the curve necessary to conform to the variation in length. Consequently in cases where portions of the boiler are rigidly secured to one another, it is necessary that the bends in the tubes should be considerable, so that they can easily, and without excessive strain, suit differences of temperature. The practice of his firm was to aim at allowing tubes freedom to expand and contract; and as the contraction and expansion of a group of tubes had been found to be nearly uniform under the conditions of working, any small difference of length seemed to be met by the elasticity of the material. Mr. Yarrow, and those who worked with him felt satisfied on this point, having made very careful experiments to test it, and he thought it would be admitted that it was most desirable, in fact almost essential, that access be freely obtained to the interior of the tubes for the purpose of examination and for cleaning. For these reasons he

was strongly in favor of straight tubes, which at the same time reduced the cost of construction, and was a convenience in actual working on account of the ease in supplying spare tubes. In this way there was avoided the large variety of forms of spare tubes necessary with boilers where the tubes were curved to different forms.

It was often asked, Mr. Yarrow continued, how, in boilers such as those in H. M. S. *Hornet* and other previous vessels, they managed to replace the tubes. The form of boilers adopted in the *Hornet*, to which the speaker was referring, was not of the type described in Mr. Milton's paper just read, and, as throwing some light on the subject, it might interest the meeting to know that on one of the trials with this vessel (in the boilers of which there are over 8,000 tubes) one of the tubes gave way on account of faulty manufacture, and the time occupied in taking out and replacing this tube ready for expanding was only 40 minutes. His firm were also led away in their first boilers with the idea that down pipes were necessary to insure circulation, and in all the boilers made by them up to three years ago they provided four down pipes, two at each end, these down pipes being suitably curved to allow for variations in lengths between the upper and lower portions of the boiler. For purposes of experiment they removed two of the down pipes, and found the action of the boiler unaffected. They then ultimately dispensed with the other two, with the same result, thereby saving both weight and length, and obtaining greater simplicity. Mr. Yarrow maintained that those who had had both water-tube boilers of a satisfactory type under their management, and those of the locomotive type, would be ready to admit that it required less skill to stoke a water tube boiler than a locomotive boiler, as with the latter it was necessary to keep the grate uniformly covered with fuel, for fear of admitting cold air, while with the former such care was unnecessary. In order to test the effect of sudden changes of temperature on one of his firm's water-tube boilers, they had, when experimenting with it in the yard, forced it excessively, and had then suddenly pulled out the fire and stripped the boiler of its casings, so as to cool it as quickly as possible. They then

replaced the casings, lit up as rapidly as possible, raising steam to 180 pounds. The result of this somewhat severe treatment showed that there was not the slightest alteration in form or any injury to the boiler. The results obtained with one of the boilers of H. M. S. *Hornet*, Mr. Yarrow thought, would, perhaps, interest the meeting. It weighed complete, with fittings, smoke box, fire doors, fire bricks, funnel, casings and all boiler mountings, also including water up to working level, 5 tons 7 cwt. On carefully conducted experiments in the yard it was found that 12,500 pounds of water were evaporated per hour from 60 degrees Fahr. to 180 pounds pressure. In order to compare the relative efficiency of water-tube boilers with locomotive boilers, there were the following facts: In H. M. S. *Havock* his firm had placed two locomotive boilers, and the machinery indicated on trial about 3,500 horse power, with an air pressure of 3 inches. In the *Hornet*, a sister ship, provided with similar engines and fitted with eight water-tube boilers, as previously mentioned, they obtained, with a very low air pressure, averaging 1¼ inches, 4,300 horse power. The eight boilers in the *Hornet* weighed 11 tons less than the locomotive boilers in the *Havock*.

They had every reason to fear, from what they had been told, that trouble would be experienced in working a group of small rapidly evaporating boilers in the *Hornet*, but as a matter of fact, they had experienced no difficulty whatever, the feed being arranged in, as it were, two stages. The feed pumps on the engines take their suction from the hot-well, and deliver into a reservoir at 50 pounds pressure, and from this reservoir the donkeys take their suction, each boiler being provided with an independent donkey. By this means a very ample supply of water was always insured on the suction side of the donkeys, and the pipes leading to them could be of moderate dimensions, in consequence of the 50 pounds pressure delivering the water readily to the suction side of the pumps. To insure the reservoir always being well filled, there was a small amount of auxiliary feed constantly passing into it, the surplus, beyond that which was necessary for the supply of the boilers, being returned to the tank through a relief valve loaded to 50 pounds. As was

well known, his firm had been strong advocates for the locomotive type of boilers, especially as the form they had adopted was found to work satisfactorily and to be durable ; but the recent success with water-tube boilers obliged them to advance with the times and to modify their opinion, a course more particularly necessary in view of the possible further rise of working pressures in the immediate future. The recent action of the Admiralty in, as it were, forcing on numerous contractors the adoption of water-tube boilers had frequently been unfavorably criticised, but Mr. Yarrow believed that time would prove the greatest possible credit was due to the authorities for having thus taken the initiative in the introduction in this country of a form of boiler which there was little doubt is destined to replace old types. He was fully convinced that, in spite of those failures which were naturally to be expected to accompany such a sudden change of practice, naval architects and engineers would, in a few years, recognize in the action of the British Admiralty a wise and important step in the advance of marine engineering, and he believed before very long that it would be generally admitted, not only in this country, but throughout the world, that the greatest credit was due to the authorities at Whitehall for having had the boldness to act as they had done.

Mr. J. I. Thornycroft said Mr. Yarrow had mentioned a case in which he had cooled down one of his boilers and afterwards got up steam again, but he (the speaker) had emptied a Thornycroft boiler and filled it up with cold water without damaging it. He further said that, with regard to dryness of steam, Mr. Seaton had remarked that he did not like a foaming boiler. The speaker did not like to say much on this point at present, but so far as he could make out, when working with his boiler and evaporating 14 pounds of water per square foot of heating surface per hour, there was less than 2 per cent. of moisture in the steam. He had also made experiments with a drowned tube boiler, and found the steam drier than he expected. He thought this dryness of steam in the water-tube boiler might be accounted for by the bombardment of the steam by small

drops of water projected through it. Each drop would form a nucleus round which would collect the smaller particles of water due to dampness, and the whole would be precipitated, leaving the steam dry to go off to the engine. The action was probably much the same as in the case of fog passing through the branches of a tree, when the mist became deposited on the twigs, and dropped to the earth as water. With regard to straight and curved tubes, there was evidently a difference of opinion. Mr. Yarrow preferred them straight; he liked them curved. The curved tubes give a greater area of heating surface within a given space, and sudden changes of temperature would have less effect. It was a mistake to suppose that cold water could get into the heating pipes, as the flow was always upwards. As to the absence of the down-comer tubes, he quite corroborated what Mr. Yarrow had said, that the boiler would work without them, for he had plugged them up as an experiment; but if they were absent it was a matter of uncertainty whether the water were going up or down, and he thought if trouble did not ultimately arise from this cause, it would be fortunate. It had been claimed in other boilers, as an advantage, that the tubes were got at by means of a door, but in his boiler there were two doors, which also had the advantage of opening inwards. With regard to manholes and openings with machine-work, which could be operated quickly, it might be well enough with new boilers, but if they had been in use for some time it would be another question.

Mr. W. H. White, speaking of the adoption of Belleville boilers for the *Powerful* and *Terrible*, said the Admiralty authorities had not decided on this course without due consideration, and had sent out an engineer to Australia to make passage in the vessels fitted with water tube boilers, in order that they might obtain full information of their working. Two years ago, when in France, he was impressed by the fact that there was not then building for the French Navy any vessel whatever, from the largest ironclad to the smallest torpedo-boat, which had not a water-tube boiler. The Admiralty, after the facts reported to

them in regard to the Australian boats, and from the experience gained by the *Speedy* and other vessels, had determined to fit the most tried types of water-tube boilers. Mr. Howden had given figures as to weight, and of space occupied by ordinary boilers; he would ask if these figures were not based on certain figures which had been previously quoted in regard to the Howden system of forced draft. Mr. Howden replied that they were. Mr. White said in that case it laid the statement open to doubt, as Mr. Howden's former figures had already been questioned, but, in any case, he did not think Mr. Howden's unfortunate experience of years ago should be taken to bias present judgment of the water-tube boiler.

Mr. Howden, after inviting attention to the record shown by the locomotive boiler on page 378, said: "Another case which illustrates the fallacy of the claims of the water tubists as to their boiler being absolutely indispensable for these fast torpedo-catcher steamers now being produced for Her Majesty's Navy, is that of the *Speedy*, of Messrs. Thornycroft, one of the latest examples of these water-tube boilers, and which has given, I understand, about the highest—if not the very highest—results yet attained. The comparative case of the locomotive boilers is that of the *Satellit*, built some time ago by Schichau, of Elbing, for the Austrian Government. These boilers, be it observed, are not worked with the closed-stokehold forced draft.

	Speedy.	*Satellit*.
Name of builders	Thornycroft.	Schichau.
Number of boilers	8	4
Kind of boiler	Water-tube.	Locomotive.
Average performance	4 500 I.H.P.	4,792 I.H.P.
Maximum performance	4,674 I.H.P.	4,901 I.H.P.
Total heating surface	15,145 sq. ft.	8,417 sq. ft.
Total grate surface	216.6 sq. ft.	185 sq. ft.
Weight of boilers, with water, uptake, funnel, &c.	90.4 tons.	96.1 tons.
Weight of boiler and water per indicated horse-power	43.32 lbs.	44.09 lbs.
Indicated horse-power per square foot of heating surface	.29	.56
Total length of engine and boiler room	101 ft. 6 in.	83 ft. 8 in.

The trials of the *Satellit* were two in number, one of eight hours' duration and one of six hours. The air pressure in ash pit

COMPARATIVE PARTICULARS OF MARINE BOILERS.

Number of column	1.	2.	3.	4.	5.	6.	7
Name of steamer	X.	*Nero*	*Polynesien*	I.	II.	*Indiana*	III.
Kind of boiler	Cylindrical	Babcock & Wilcox.	Belleville	Cylindrical	Cylindrical	Cylindrical	Cylindrical
Number of boilers	4 D.E.	1	20	3 D.E.	2 D.E.	1 S.E.	8 D.E.
Diameter and length of boilers	13 ft. 1½ in. × 18 ft. 7½ in.	.....	7 ft. × 7 ft. square.	14 ft. 6 in. × 19 ft. 6 in.	15 ft. 6 in. × 20 ft.	15 ft. × 11 ft. 9 in.	16 ft. 4½ in. × 20 ft.
Number of furnaces	24	.....	20*	18	12	3	64
Heating surface, square feet	16,000	2,272	23,800	1,400	10,000	2,338	42.500
Grate surface, square feet	560	44	580	360	250	56.5	1,280
Weight of boilers with water, uptakes, funnel, and mountings, including all forced draft apparatus	364 tons	41.65 tons	380 tons	361 tons	272 tons	89.75 tons	1,019 tons
Space occupied on floors by boilers and stoke holds	1,904 sq ft.	.....	2,100 sq. ft.	1,597 sq. ft.	1,152 sq. ft.	314 sq. ft.	4,896 sq ft.
Space occupied on floors by boilers only	.....	.....	980 sq. ft.	877.5 sq. ft.	640 sq. ft.	182 sq. ft.	2,720 sq ft.
Indicated horse-power on trial	9.400	530	8,000	8,500	6,000	1,400	25,000
Economical indicated horse-power at sea	.....	450	5,000	7,500	5,000	1,300	21,000
Ratio of grate surface to heating surface	$\frac{1}{28.5}$	$\frac{1}{51.6}$	$\frac{1}{41}$	$\frac{1}{38.8}$	$\frac{1}{40}$	$\frac{1}{41.3}$	$\frac{1}{33.2}$
Heating surface per indicated horse-power on trial	1.7	4.28	2.97	1.64	1.66	1.67	1.7
Heating surface per indicated horse-power at sea	.....	5.04	4.76	1.86	2	1.79	2.02
Indicated horse-power on trial per ton of boilers, water, &c.	25.82	12.72	21.05	23.54	22.05	15.59	24.53
Indicated horse-power at sea per ton of boilers, water, &c.	.....	10.8	13.15	20.77	18.38	14 48	20.6
Indicated horse-power on trial per square foot grate surface	16.78	12.04	13.79	23.61	24	24.77	19.53
Indicated horse-power at sea per square foot grate surface	.....	10.22	8.62	20.83	20	23	16.41

* Furnaces in these boilers are of extra large size, and require two furnace doors for each.

was from three inches to four inches water gauge. The boilers, on examination after trial, were absolutely faultless. The vessel, after trial proceeded from Elbing to Pola, in Austria, crossing the Bay of Biscay in midwinter. The boilers were found in perfect order on arrival after her long voyage, and she was immediately put in commission. These particulars I have from Herr Ziese, managing director of the Schichau establishment, who vouches for their correctness.

He also submitted the preceding table, which called forth the remarks of Mr. White, previously quoted, and the correctness of which has been questioned in many quarters.

The following are the explanations of the different columns:

No. 1 column contains the particulars of the boilers of a cruiser belonging to a certain government which it is not necessary to specify. The boilers are constructed in the style of those of our Admiralty, and they are worked with the same forced draft. The weights were carefully and correctly taken. The indicated horse power given was that obtained on a four hours' full forced-draft trial. The tube ends on this trial suffered some injury.

No. 2 is the Babcock and Wilcox boiler fitted in the steamer *Nero*. The weight of this boiler in the Table I, exhibited at the meeting, given as "over 40 tons," was called in question by Mr. Rosenthal, representing the Babcock and Wilcox boiler, who said that with steam up the *Nero's* boiler was under 30 tons weight. Though I had made considerable efforts to obtain the exact weight of this boiler with its fittings, and had sent a special representative to Hull for this purpose a considerable time before the date of the meetings, I had only then obtained the following particulars:

	Tons.	*cwt.*	*qr.*	*lb.*
Total weight of boiler, with water and casings, but without funnel and mountings	27	8	1	11
Weight of fresh-water tank and water, fitted with boiler	9	0	0	0
Making together	36	8	1	11

But in addition to the funnel and mountings not given, there was the reduction of the fire grate, by means of fire brick, from 57 square feet to 44, besides fire-brick slabs being put on the

tubes to give a zig-zag circulation to the gases, all which was going on while my representative was at Hull. I felt sure that all these items would be more than 3 tons 11 hundreds 2 quarters 17 pounds required to bring the weight up to 40 tons. Within the last eight days Mr. Spear, the owners' superintendent, has informed me that the additional weight of the funnel, &c., is 4 tons 5 cwt., and that the additional brick work used in the alterations on furnaces will not exceed a ton. This, therefore, makes an addition of 5 tons 5 cwt., which, added to the 36 tons 8 cwt., makes 41 tons 13 cwt. as the total weight of this boiler, with water and mountings, as now fitted in the ship. It is but right to mention that Mr. Spear says: "In regard to the fresh-water tank, probably after further experience we might be able to dispense with it." When that time arrives, and it is found that the boiler can run safely without it, I would like the fact to be reported; but as this tank is meantime an integral part of the boiler fittings of the *Nero*, and, as in the other cases given in the table, all mountings and fittings whatsoever are included in the weights, and, in the forced-draft cases, the fans and their engines, with steam and air piping, as being necessary adjuncts, it is clear that this tank must meantime form part of the weight of the *Nero's* boiler and fittings.

[**Mr. Rosenthal,** the representative of the Babcock and Wilcox Company, said: "This tank was not fitted by ourselves, and in any case is no more a necessary accompaniment of the water-tube boiler than are the hundreds of tons of fresh water which are sometimes carried in the double bottoms of ships, and which surely he would not consider as part of the weight of the boilers.

["I would only add that in the design of the *Nero* the question of weight was not studied at all, and it is very much in excess of what we should fit for a boiler of similar power if we were desired to restrict the weight. I have also to state that the funnel and casing were made fully 30 per cent. heavier than usual, at the desire of the owners."]

Column 3 contains the particulars of the Belleville boilers in the Messageries steamers *Polynesien*, *Armand Béhic*, and others belonging to this company.

No. 4 column gives the particulars of three double-ended boilers, made to Board of Trade rules, for 175 pounds working pressure. The dimensions of shell for heating surface are what I recommend for steamers requiring a fair power per square foot of grate surface and heating surface, without the extra size and weight of shell, which conservative owners and superintendents mostly, as yet, desire even when working the boilers at a very moderate forced draft power.

No. 5 column gives the dimensions and particulars of two double-ended boilers, calculated and proportioned in the same manner to supply steam economically at sea for 5,000 indicated horse power.

No. 6 column gives the particulars of the boilers of the *Indiana* and sister ships, as already mentioned. These boilers have been running regularly for several years between Philadelphia and Liverpool or Antwerp, and have developed the powers stated in their voyages across the Atlantic. These boilers are considerably larger and heavier than is required for the size of the furnaces and heating surface, and have large water spaces. The water alone weighs 26½ tons. It should also be mentioned that the weights of the cylindrical boilers I, II and III have been very carefully calculated and checked, and include every item connected with the complete installations.

[On page 471 of Volume III of the JOURNAL will be found some particulars of the machinery of the *Illinois*, a sister ship to the *Indiana*, with copies of indicator diagrams taken on her official trial. The boiler is, apparently, identical with that of the *Indiana*, and the I.H.P. shown by the cards, which were undoubtedly good ones, is 1,257.—EDITOR.]

No. 7 column gives particulars of boilers of 25,000 indicated horse power, calculated on Admiralty scantlings for shells. These boilers are given here as of equal indicated horse power to the proposed Belleville boilers for the *Powerful* and *Terrible*, and at 25,000 indicated horse power would be worked on a consumption of about 1.5 pounds to 1.6 pounds per indicated horse power per hour, according to the size of the engines.

In considering the ratios of indicated horse power to heating surface and grate surface of the cylindrical boilers I, II and III, it will be noticed that they are less than that of the *Indiana* and sister ships, so that the economical indicated horse power to be obtained from these three several installations is assured from the fact that fully more has already been obtained from cylindrical boilers at sea under less favorable circumstances than would now be brought to bear on such forced-draft cylindrical boilers as here specified, for it is to be understood that all these installations are taken as worked under my system of forced draft.

In comparing the sea-going water-tube boilers of this table, the *Nero* and *Polynesien*, it will be seen that the *Nero's* boiler is certainly not of a kind fitted for sea-going purposes. The indicated horse power obtained at sea, which I have stated at 450, though in the last statement I received it was given as 440 indicated horse power, is a power far too low for the heating surface, weight and size of boilers to make it suitable for steamships. I understand also that the consumption with even over 5 square feet heating surface per indicated horse power is considerably over that required for cylindrical boilers.

In the Belleville boilers, though the trial indicated horse power is given at 8,000 and the heating surface even at that high power is not far from double that of the cylindrical boiler I, in column 4, the consumption of coal, I am convinced, could not be less than 30 or 40 per cent. more than that of this cylindrical boiler installation when working at the same power. It will be seen, therefore, that the cylindrical boilers I, which occupy only .76 of the space, and only .95 of the weight, are not only more powerful, but would be in a vast degree more economical.

As the ordinary sea working of the Belleville boilers is 5,000 indicated horse power, installation No. II of the cylindrical boiler would be quite equal to them working at this power, and I am certain would be also much more economical. It will be seen that these boilers weigh 108 tons less, and occupy a space of only .55 that of the Belleville boilers.

[It was stated that the *Armand Bêhic* uses only 19 of her 20 boilers, and that, therefore, Mr. Howden's figures needed correc-

tion accordingly, *i. e.*, the grate surface would be 551, the heating surface 22,610 square feet, and the boiler weights 361 tons.]

I am not aware of the dimensions of the Belleville boilers proposed to be fitted in the *Powerful* and *Terrible*, but I say that if they are to be capable of keeping the sea at a high speed the boilers cannot be much less in weight and in the space they occupy per indicated horse power than the Belleville boilers of the *Polynesien* and sister ships.

Though almost every possible merit is at the present moment being claimed for the water-tube boiler, I have not yet heard of any advocate so bold as to claim that it is more durable than the cylindrical boiler. Mr. Milton, in his Cardiff paper of last year, mentions several small French steamers which had used water-tube boilers for a considerable number of years, but of the work which these boats have done, or the number of times these boilers have received new tubes or renewals of parts, no mention is made. It is well known that the tubes, which in every design form the principal part of the boiler, are frequently renewed. In most of the designs, and very noticeably in the Belleville design, this idea of providing for frequent repair is so prominent that it appears to be its dominant feature, and I think very wisely too. The feature also of great subdivision in these boilers arises entirely from this idea of frequent repair being necessary, further provision for it being thus made, as well as in the design of the boiler itself. The multiplication of boilers gives the opportunity of making repairs at sea without losing a large proportion of the boiler power when such repairs are being made. This I have heard stated frequently by its advocates as one of the advantages of the Belleville boiler. Since writing my first letter the following paragraph has appeared here from the Londen correspondent of the "Glasgow Herald":

"Much has been made of the breakdowns of the torpedo gunboat *Sharpshooter* in her trials of the Belleville water-tube boiler, but however annoying these are to the parties interested, they do not awaken surprise at the Admiralty. In the first place, it is always difficult to adopt new appliances under old and prescribed conditions, and, moreover, the breakdowns have

not all been associated with the water-tube boiler. The difficulty is the working of the automatic feed-gear. The water evaporation is so rapid and great that there is always present the liability to get out of gear. But the work of an hour or two puts it right. To meet this, I understand, French mercantile vessels have as a rule 10 per cent. more boiler power than they require, so that when anything goes wrong with one boiler there will always be a reserve boiler for use, and I learn that in the new cruisers *Terrible* and *Powerful*, building respectively by Clydebank and Barrow firms, there is even a greater than 10 per cent. ratio of reserve boiler power, so that even should the feed gear to any boiler fail to act, it can quickly be repaired, and the steaming efficiency of the vessel will not in the meantime be affected. The facility of repair is a most desirable feature, and no danger is involved in the breakdown of the gear."

The above no doubt gives the current talk of those in touch with our naval circles. When Mr. Milton, therefore, mentions that a certain water-tube boiler in a steamer is found, after some years' working, to be in good condition, it should also be explained how much of the original boiler is left, and how often the tubes have been renewed.

There are many cylindrical marine boilers to-day doing duty as well after over twenty years' constant work as they did when new. I may point, for example, to the original White Star Atlantic steamers running yet with the same boilers, and doing as good work, if not better, than when they started over twenty years ago. No doubt they have been repaired in some places and re-tubed, but the main structure is the same.

I may instance the case of the *St. Magnus*, of the North of Scotland, Orkney and Shetland Steam Navigation Company, which I completed with new engines and boilers in February, 1875. These boilers, now in their twentieth year, still carry the original pressure of 70 pounds, and, from inquiry made a week or two ago, the company's manager informs me that there is not a patch or repair on any furnace, combustion chamber, or outside shell—that is, no repairs for over nineteen years' work, the tubes only having been renewed. The treatment has simply been

that of fair attention when running, and frequent changes of water. As the steamer is laid up in the winter months—a condition often more injurious to boilers than working—the interiors are then washed out and dried, and no moisture allowed to gather on the plates while lying up.

To show how cylindrical boilers are equally fitted for forced-draft working, and especially in regard to durability of tubes, which are supposed to suffer by forced draft, and do suffer greatly by closed-stokehold draft, I may instance the case of the boiler of the first steamer I fitted, the *New York City*, which has now been running constantly for nine and a half years. On inquiry at the owners' within the last few days, I find that not a drop of leakage has ever been found at a tube end, and that all the original tubes, with the exception of one, remain in good condition after nine and a half years' work. The single tube removed was found to be chafed by the friction of the hanger of a zinc plate, and was therefore replaced.

The Clan Line have now twenty-two steamers running with cylindrical boilers and the writer's forced draft, beginning in 1888. The company, within the last few days, have informed me that leakage of tubes or other troubles are unknown in these steamers; some of them, making continuous voyages of 23,000 miles in length, have frequently had occasion to be sent off in three days after arrival in port on a voyage of similar length. There can be no repairs necessary in such boilers.

The report mentioned by Mr. F. C. Marshall, of what a personal friend had witnessed in Toulon and the South of France, the home of the water-tube boiler, that all ships lying there fitted with such boilers were having their tubes taken out, shows the constant repairs and renewals which these boilers require.

It is surely unnecessary to say more in order to prove that the cylindrical boiler is as superior to the water-tube boiler in durability as it is in efficiency, and that the cost of maintenance must be in proportion to the durability and absence of renewal of parts.

It is very generally supposed that water-tube boilers are safer than cylindrical boilers because of the individual parts be-

ing able to stand a very high cold-water pressure. This idea is, however, quite erroneous. The safety of a boiler depends on its homogeneity, both in structure and contained water, the quantity of water contained, the equality of its strains, the thickness, within limits, of its material, its capability to receive and dispose of the greatest possible furnace heat most rapidly, and in having the fewest number of separated or detached parts.

A leakage in a tube of a cylindrical boiler is an occurrence of no great consequence, such leakage generally arising from pitting, and begins with a very small hole, for which it is seldom necessary to stop the ordinary work of the boiler, as the tubes can be plugged so as to prevent leakage until a new tube is put in, or a proper tube stopper fitted, if no tube is at hand. As the pressure of the water is on the outside of the tube in a cylindrical boiler, the tendency is to prevent the rupture of the tube, while in the tubulous boilers, with the pressure *inside*, the tendency is to cause a rupture. A leak in a single water-tube causes the stoppage of the boiler.

Regarding economy of fuel, I believe no one who has had any experience of cylindrical and water-tube boilers will fail to admit that the water tube falls considerably behind. The reasons must be obvious to those who have read my paper. It is a defect arising from the very nature of the water-tube boiler. The principles referred to in my paper show that by using judiciously a large heating surface of small and consequently short-lived tubes, the water-tube boiler may be made much less wasteful in fuel, and be capable of being worked with forced draft for the time the tubes may hold out; but even these boilers are much less economical than the cylindrical. For large water tubes capable of running at sea for a considerable period, the same principles will show that a high rate of combustion per square foot of heating surface must necessarily be most wasteful in fuel. From a long experience and study of both cylindrical and water-tube boilers, I have no hesitation in deliberately saying that, compared with a properly worked cylindrical boiler, having one-half the tube heating surface of Belleville boilers, such as in the Messageries mail steamers, the latter would, at a moderate rate of com-

bustion per indicated horse power or per square foot of heating surface, require 25 per cent. more fuel per unit of horse power, and at a fairly high rate of combustion they would require from 30 to 35 per cent. more than the cylindrical boiler.

Mr. Milton gives some cases in his Cardiff paper on water-tube boilers, having, it is understood, tubes $2\frac{3}{4}$ inches in external diameter, of what appears to be a fairly economical rate, but on examination it is found that the rate of combustion in the most economical case, in which the actual evaporation is 10.66 pounds of water per pound of Cardiff coal, and 12.41 pounds from and at 212 degrees, is only 10.28 pounds coal per square foot of fire grate, and 3.06 pounds evaporation per square foot of heating surface. When the combustion was increased to the very moderate rate of 15.38 pounds per square foot of fire grate, and the actual evaporation to 5.50 pounds per square foot of heating surface, the evaporation per pound of coal fell to 8.96 pounds, or 16.7 per cent. less, and when the combustion was increased by fórced draft to barely 25 pounds per square foot of grate and the evaporation to 6.69 pounds per square foot of heating surface, the evaporation per pound of coal fell to 8.01 pounds actual, or 24.77 per cent. This altogether proves my contention that the water-tube boiler cannot be worked at any high power without sacrificing fuel greatly. At extremely low rates of combustion, so low that no shipowner could afford to carry boilers of the size and cost for the trifle of work got out of them, such boilers show a fair economy, but long before they are worked up to an ordinary working rate the economy disappears, and consumption per indicated horse power rises rapidly.

As I have shown that the heating surface of the water-tube boiler is much less effective than that of the cylindrical, the low rate of combustion gives the water-tube an immense heating surface to take up the comparatively small heat, and accordingly the fuel combustion is affected advantageously, and there is comparatively little heat lost by radiation and by the waste gases, hence the economy. When the rate of combustion is much increased, the inefficient and wasteful character of the boiler reveals itself. It must be remembered that the loss by radiation

in these boilers is often very great. In the cylindrical boiler it must be remembered the heat of the outside plating, which requires to be protected, can never be more than the temperature of the steam inside whatever rate of combustion is used, say about 270° for 160 pounds pressure. This is easily dealt with by good non-conductors. It cannot, therefore, be a matter of surprise that at high rates of combustion the water-tube boiler, fitted for seagoing steamers, is a wasteful boiler.

Mr. Milton in his recent paper mentions trials of Messrs. Anderson and Lyall's combined boiler, where with Welsh coal and a combustion of 18.29 pounds per square foot of fire grate, the equivalent evaporation from and at 212° was 12.05 pounds. But as neither the grate surface nor the heating surface is given, whether this result is good or bad cannot be known.

Notwithstanding the fact that the French mail steamships use large water tubes only, I am informed that our Government ships are not to be fitted with this sea-going water-tube boiler, but with the war ship kind, which have much smaller tubes. Now, this kind of water-tube boiler has not to my knowledge been tried but in war ships, and therefore any experience derived from the Messageries mail steamers is of no direct use. It is after all a practically untried boiler they are adopting. It will not be argued that the trials of the *Speedy* and such craft give any guidance for sea-going steamers.

I now come to the consideration of Dr. White's reasons for our Admiralty authorities adopting French water-tube boilers for the most powerful steamships of our Navy. These were, shortly, (1) the fact that all the steamers of the French Navy, large and small, were now having water-tube boilers; (2) the experience gained from the performances of the *Speedy* and other fast small steamers with water-tube boilers built by Messrs. Thornycroft and Mr. Yarrow; (3) the Admiralty authorities had before deciding on adopting water-tube boilers sent an engineer to Australia in the mail steamers of the Messageries Maritimes Company, in order to obtain all information about their working.

When the engineer from the Admiralty went one voyage to Australia to see these boilers at work, he appears to have been

struck with the fact that they could run continuously without breaking down, but, had he gone for ten years to Australia in our own ships without ever seeing a breakdown, he would probably not have been surprised, because that is merely what he would have expected.

The fact that the French Admiralty have gone in for water-tube boilers in their ships is easily explained. Some years before our Admiralty, they began to fit their warships' boilers with closed stokehold forced draft, and some of their boiler designs, which I have come to know of, were even more unsuited than those of our Admiralty for this injurious and wasteful system. The French Government, though testing their warships even less, I believe, than our Admiralty, saw and acknowledged their mistake sooner, and evidently thinking that these injuries arose from something inherent in the cylindrical boiler, and finding water-tube designs by their own countrymen which, when tried, did not leak at the tube ends, they at once began to use them exclusively. The French are idealists and patriotic, and as they found they could apparently do better with the new boiler than with the badly designed cylindrical boiler worked with an injurious forced draft, they adopted the water-tube boiler from a patriotic sentiment as well as from a belief in its merits. The highest subsidized company in the world, the Messageries Maritimes, also tried these boilers, and, as is well known, have fitted the Belleville boiler, designed for working at sea, to a number of ships, and have worked them in a very admirable manner.

Mr. Sampson made some fun over the remark in my paper, that if the automatic feed for the Belleville boilers was so perfect that one man, unassisted, could, with it, tend a ship with twenty boilers, this automatic feed regulator must be a marvel, and the attendant a specialist of a very high order. Mr. Sampson said the man *was* a specialist; he was an Arab. This fact explains a good deal, for all engineers in command of steamship machinery know that if they want orders to be strictly carried out to the letter, they find none who will do this so well as Orientals after they have been properly instructed in the duty required of them.

It therefore shows the wisdom of the French company in having such men to attend to the regulation of the feed water in their Belleville boilers. Mr. Sampson, however, not content with this score, went further, and in order to show how very perfect this automatic feed was, and also the little importance of having the water at any particular level in the Belleville boiler, said it did not require any attention, as it took care of itself, and even water-gauge glasses were not necessary, as the engineers in these ships were so assured of safety that the last thing they thought of was the water level.

Mr. Sampson said: With regard to the automatic feed of the Belleville boilers, the efficiency of which Mr. Howden so persistently misbelieves, I can only repeat that feeding the boilers gives much less trouble, if any trouble there be, than with the ordinary marine type, and, moreover, although Mr. Howden refuses to take as a serious statement that vessels run at sea without gauge glasses, if he will take the trouble to come with me to Marseilles I will give him an opportunity of making a trip upon one of the ships in question, where he can witness facts for himself. If Mr. Howden were to ask the engineer officer who was appointed by the Admiralty to report upon the working of these boilers, he would learn that upon a voyage of 10,000 miles no gauge glasses were used, the engineers preferring to work without them.

Now, to pass to the question of H.M.S. *Sharpshooter*. The *Sharpshooter's* tubes did not leak either upon the trials at moorings or at sea. It was some of the feed pipes which gave trouble, and it was nothing whatever to do with Messrs. Belleville or their boilers. The other breakdowns reported were due entirely to derangements with the engines. The automatic feed did not become deranged, but a stoker who had cleaned the apparatus omitted to connect the float to its spindle, the error not being discovered until steam was raised; but the day was not wasted; it was very speedily put in order, and the vessel finished her trials. If the connecting-rod is not coupled one can scarcely expect the cranks to turn.

I can also positively assert that in the case of a set of Belleville boilers fitted by my firm to a Russian coast defence vessel, from the first steam at moorings to the completion of the guarantee period of one year, not one penny piece was spent either upon adjustments or maintenance of the boilers. Not a single leak or trouble of any sort was experienced, in connection with Messrs. Belleville's apparatus. The machinery during the trials was in charge of and manipulated by the usual staff of Russian artificers and stokers, no "specialists" being employed, and all will bear the strongest testimony to the efficiency of the boilers and the facility with which they can be worked.

When I alluded to the leading stoker being an Arab, I certainly intended no disrespect to Orientals, whom I know from experience at sea to be most useful men in the machinery department; but Mr. Howden insinuated that for twenty boilers we should require twenty high-class specialists to manipulate the feed, which would be, to say the least of it, somewhat droll.

It is very easy to make misleading statements and to put the same in print, but whilst carefully suppressing the names of the vessels carrying his boilers, he gives the names of those with water-tube generators, at the same time vastly underrating their performances. For instance, the mail steamers *Australien*, *Polynesien* and *Armand Béhic*, each fitted with twenty Belleville boilers, on voyage between Marseilles and Australia, indicate 5,000 horse-power, with sixteen boilers in work out of twenty—burning briquettes and Australian coal. They do not use more power, because they can very well keep time, steaming easily with the reduced number of boilers, but if another knot be required to compete with a rival line, it is there on board to be had at a moment's notice. Let Mr. Howden tell us to what vessels these phantom boilers are fitted which, at an average economical sea speed, would indicate 22.22 horse-power per square foot of grate, and require only 1.75 square feet of heating surface per indicated horse power. Now, sir, I have invited Mr. Howden to visit Marseilles with me and be convinced; will similar courtesy be extended upon his side by inviting me to make a voyage upon the vessel fitted with boilers as per his list

of comparisons? If he does so, and proves to me that upon a voyage of, say, six days a mean of 8,000 horse-power is produced from boilers having 360 square feet of grate, and 14,000 square feet of heating surface, and weighing with water and all fittings 361 tons, then he will have no stronger advocate for his system than myself.

Mr. C. Humphrey Gilbert said: The paper gives the weight of boilers capable of maintaining a continuous sea power of 8,000 horses as 361 tons, including water uptakes and funnels. Mr. Milton cited the P. and O. R.M.S. *Himalaya*, a vessel fitted with boilers worked on Mr. Howden's system, as being of about that power, and quoted (from Messrs. Maudslay's book) the weight of her boilers, including water, but without uptakes or forced draft appliances, as 546 tons, grate area and heating surface being given as 496 and 20,940 square feet respectively. Now, assuming these figures to be correct, one cannot help asking why Mr. Howden has not been able to obtain better results for the P. and O. Company, one of his best customers.

There is not a large excess of boiler power in the ship mentioned; there is certainly no more than is necessary for 8,000 indicated horse-power. As one who has had some stokehold experience on board of her, I can vouch for it that she is by no means always easy to steam. Moreover, the figure 8,000 does not represent her continuous sea power; on her trial she exceeded this figure by 300 horse-power, but in ordinary running at sea, even when "driving," 7,000 to 7,500 is all she does; this, again, of necessity represents the power at the time indicator cards are being taken. All sea-going men know what this means; for a few minutes every morning all auxiliary machinery is shut off, so that the main engine may get every pound of steam; every exertion is used to run up the boiler pressure within an ace of blowing off, and an extra revolution or two is got out of the engine, all to enable the chief to get "a good card." It is plain, then, that the *Himalaya* does not exceed 7,000 I.H.P., continuous sea steaming, and even this power is not always easy to keep up. With at all indifferent coal, it is only during the mid-

dle hour of the watch that it is possible to keep the steam at or near 160 pounds; often enough while fires are being cleaned, or towards the end of the watch, these Oriental firemen, in whom Mr. Howden places such faith, require coercion of a very forcible kind to keep up the steam. Frequently the regulation ½-inch air pressure below the grate becomes more like 1¼-inch or 1½-inch, the pressure in the air trunks leading directly from the fans often blowing the water clean out of the gauges, although they register up to 4½ inches.

In other words, 496 square feet of grate, and 20,940 square feet of heating surface, are not more than enough for a continuous 7,000 I.H.P., which alters Mr. Howden's figures from 22.22 to 14.11 I.H.P. per square foot of grate, and from 1.75 to 3 square feet of heating surface per horse-power. Perhaps these results might be improved upon with good coal and English firemen, but Mr. Howden is making a comparison with French ships on the same route, using the same coal and carrying native firemen.

A correspondent of "Engineering," under the *nom de plume* of "Automatic," writes: The remarkable results obtained with Mr. Howden's boilers on merchant steamers most certainly entitle him to an attentive hearing on that subject, concerning which he ought to know all that is worth knowing; but why not leave the defense of locomotive boilers to those who build them? Many engineers in England would be able to speak with authority on the subject, and Herr Ziese's name is no stranger. In any case I doubt whether the well-known engineer who designed the boilers of the *Tripoli*, would have claimed for them the astonishing performance put forward by Mr. Howden, who assumes that with 28 square feet of grate area one of these boilers supplied steam for 1,120.86 indicated horse power, or 40 horse power per square foot of grate. The Italian Navy would have reason to be proud if Mr. Howden's figures were correct; however, they are not. The *Tripoli* has six boilers, and her engines were intended by their designer to develop 4,200 indicated horse power. On trial they averaged a little more than half that power, and Mr. Howden must be satisfied with 375 for each of these boilers, or only

13.4 indicated horse power per square foot of grate. Thus Mr. Howden's comparison turns against his contention.

Turning to the other case advanced to prove the immense superiority of the locomotive boiler, it may be of interest to see what other engineers have done in the same direction, and, as the weight of boilers with mountings, fire bars, uptakes, funnels, &c., complete per indicated horse power, appears to be the most suitable term of comparison for boilers intended for extremely rapid vessels, I append the following:

	Pounds per indicated horse power.
Speedy, according to Mr. Howden	75.39*
Satellit, according to Mr. Howden	44.09
Water tube boilers now building	35.
Havock, locomotive	34.
Hornet, water tube	22.3

*Corrected to 43.32.

Perhaps Mr. Howden had not these figures at his disposal when preparing his paper, but anyone can see that they entirely and utterly demolish his contention in favor of the locomotive boiler.

Having seen what Mr. Howden can do with a locomotive boiler, let us see what he can do with a cylindrical one. In his table of comparative particulars of marine boilers, No. 1 is a double-ended boiler fitted in a cruiser of a certain unknown Government, with forced draft and closed stokeholds. No. 7 is also a double-ended boiler, and, like the former, is supposed by Mr. Howden to comply with Admiralty requirements. For Mr. Howden these Admiralty requirements have evidently remained what they were six or seven years ago, and though it may favor his case to work out boilers lighter than those of the *Medea* or the *Blenheim*, I doubt whether the Admiralty will accept his tenders when submitted for approval, as in the Navy double-ended boilers and light scantlings are things of the past. The indicated horse power per square foot of grate estimated for No. 7, *i. e.*, 19.53 on trial and 16.41 at sea, is much smaller than in the case of Mr. Howden's other vessels, so that I presume he finds the latter rates rather difficult to maintain. Perhaps the

explanation of all these things may be found where Mr. Howden puts the coal consumption on forced-draft trial at from 1.5 pounds to 1.6 pounds; no doubt that would be welcome news to many, but I fear it only proves that Mr. Howden's acquaintance with naval engines, and their little peculiarities, is neither profound nor extensive. Of No. 2, steamship *Nero*, I shall say nothing; the boilers are, I understand, of an experimental nature, and appear very heavy for their power. No. 3 gives the Belleville boilers of the *Polynesien* and her sister ships, and I think Mr. Howden might have altered his figures in compliance with Mr. Sampson's statement, viz: that 5,000 horse power are indicated at sea with only sixteen boilers. The figures for the sea performance would then read as follows.

	Sq. ft.	Instead of—
Heating surface per indicated horse power at sea	3.8	4.76
Indicated horse power at sea per ton of boilers, water, &c	16 44	13.15
Indicated horse power at sea per square foot of grate area	10.73	8.62

Nos. 4 and 5 are the *Y Z* of the table exhibited at the meeting, and are, if I understand Mr. Howden rightly, hypothetical cases, not actual steamers. As Mr. Howden's system of forced draft has been fitted on a considerable number of vessels, it appears perfectly fair to expect him to give us results actually obtained, and not to introduce the contentious element of deferred hope where it is not necessary. Why, instead of the ethereal *Y* and *Z*, does he not favor us with the concrete *Himalaya* and *New York?* No. 6, the *Indiana*, certainly presents the desirable and necessary feature of actual existence, and we may compare her with her neighbors. Strange to say, in her case, though she is somewhat of a tramp, the ratio of sea power to trial power is much higher than in the 4, 5 and 7, or in the *Himalaya*, according to another correspondent. It is true that Mr. Howden does not state that she *averages* 1,300 indicated horse power at sea, but only that she has developed that power at sea, and there may be a wide margin between the two. The *Polynesien*, I take it, averages 5,000 indicated horse power with sixteen boilers on her long journey to Australia. The horse power claimed to be

developed per square foot of grate surface on trial and at sea by the *Indiana*, viz: 24.77 and 23 respectively, calls for remark. Here she fairly beats the record, and Mr. Howden is to be congratulated, whatever his coal bill may be.

The horse power per ton of boilers, water, &c., indicated by the *Indiana*, viz: 15.59 on trial and 14.48 at sea, does not seem to compare very favorably with 21.05 and 16.44, the corresponding corrected figures for the *Polynesien*.

Another correspondent, Mr. Charles de Grave Sells, writing on the subject of the locomotive boilers of the *Tripoli*, says that the figures given in the table, page 378, can hardly be accepted as an "actual example," that the actual horse power was 580 instead of 1,120, and adds:

A more unfortunate selection for an example of the advantages of locomotive boilers could hardly have been made, for these very boilers quoted (of the *Tripoli* and *Folgore*) are to be removed to make way for water-tube boilers.

NOTES.

THE BELLEVILLE BOILERS OF THE *SHARPSHOOTER*.

On page 202 of the last number of the JOURNAL, mention was made of the fact that this vessel was to be fitted with eight Belleville boilers, and that the combined grate and heating surface would be respectively 269 and 7,696 square feet. During February and March, they were subjected to a number of tests, to determine their acceptance, the Belleville Company, however, assuming no responsibility for the I.H.P. of the engines, the condition being that, under natural draft the boilers should be capable of burning a certain quantity of coal per square foot of grate, and that the evaporation per pound of coal should not fall below a certain figure. Each set of boilers was tried separately, on different days for a period of twelve hours, during which the mean rate of combustion was 23 pounds per square foot of grate, one-half pound more than stipulated. Then followed an evaporative test, lasting eight hours, resulting in a mean of 8.2 pounds of water per pound of coal. During all of these tests, as well as another of two of the boilers under forced draft, burning thirty pounds of coal per square foot of grate, the boilers worked satisfactorily, the only hitch occurring in one of the subsequent trials, where there was trouble with the automatic feed regulator owing to failure of one of the attendants to connect the float.

The *Sharpshooter* then made three runs at sea, the first of 24 hours duration, using six boilers, being for economy, burning less than ten pounds of coal per square foot of grate. The results were: 14.5 knots, and 1,282 I.H.P. on a coal expenditure of 1.96 pounds per I.H.P. The next trial was made with one set (four) of boilers, and lasted twelve hours, the rate of combustion for one-third of the time being about 31 pounds per square foot of grate, and for the remainder about 25 pounds. The results were:

	First four hours.	*Eight succeeding hours.*
Steam pressure in boilers	186.	182.
Vacuum	28 4	28.6
Revolutions	206.9	185.6
I.H.P	1,497.	1,235.
Speed	15.2	14.0
Coal per I.H.P	2.78	2.66

The last trial was made under similar conditions with the other set, the result being:

Steam pressure in boilers	202.	170.
Steam pressure at engines	139.	124.
Vacuum	28.	28.1
Revolutions	180.	177.4
I.H.P	1,421.	1,191.
Speed	14.8	14.
Coal per I.H.P	2.93	2.78

LIQUID FUEL IN COMBINATION WITH COAL.

An experiment was recently tried on board the French cruiser *Surcouf* of burning petroleum refuse in the same furnaces with coal. The method followed was practically the same as that tried on some Italian ships last year, the furnace fronts being arranged for oil burners without interfering at all with the furnace doors, and steam being used to spray the oil, which was stowed in a tank on the upper deck. The details of the experiment are meagre, and the only information attainable is that when burning coal alone the ship made eight knots, but that with coal and oil combined she made thirteen and a half. Just how much of this additional speed is due to the additional coal burned by reason of the increased draft induced by the method of spraying the oil, and how much to the oil, can not be stated on account of the few data to be had.

GAS ENGINE FOR A SCREW SCOW.

Le Yacht gives a brief account of a scow whose propeller is to be driven by a gas engine. The service of the scow is on the Seine between Rouen and Paris, and the gas will be obtained at

St. Etienne, one charge being enough for the round trip from Rouen to Paris and return. The designer of this system, M. Capelle, believes that it is cheaper, under the special circumstances, and lighter than steam machinery. The gas, which is the ordinary illuminating gas, will be stored in heavy iron tubes side by side under the upper deck beams, under a pressure of from 100 to 160 atmospheres. The gas engine is of the two-cylinder type, rated at 40 H.P., but capable of greater power if necessary. It is placed at the stern with the fly wheel forward of the cylinders. The screw has reversible blades.

THE *BRANDENBURG* CATASTROPHE.

A dreadful accident occurred on board this German battleship, on the 16th of February, in which forty-five men were killed. She was under way making her last official trial, and at the time of the accident was developing about 7,000 I.H.P.; on one of her previous trials she had run up to about 10,000. The magnitude of the catastrophe can be better understood by a short statement of the arrangement of her steam pipes. In the main steam pipe in each engine room there is a separator, and between the separator and the high-pressure steam chest a length of copper pipe of about 10 feet, one end of which is fastened to a stop valve on the separator and the other in a stuffing-box slip joint attached to the stop valve on the steam chest.

The stop valve on the separator broke short off close to the flange, thus giving a full opening to the escape of steam from the twelve boilers. The door between the two engine rooms was open, so that the steam filled both, and of the men in them 39 were killed outright and 9 so badly injured that six of them died subsequently.

The accident has been attributed to insufficient metal in the casting of the valve chamber.

Two causes have been assigned for the accident: one that there was not enough metal in the casting, or that it was an imperfect one, and the other that the pipe pulled out of the stuffing box.

German papers have given little or no information on the subject, and the account here given is from "Mittheilungen aus dem Gebiete des Seewesens."

LIQUID FUEL AT SEA.

From the "Engineer," London, April 6, 1894.

It has often been asked why it is that steamers carrying some thousands of tons of petroleum in bulk should fill their bunkers with coal? Although it is true that petroleum in any but the crudest form is an exceedingly expensive fuel if purchased in England, it is not necessarily costly if bought on the Black Sea or in the United States. Indeed, unless a steamer carries coal enough for the voyage out and home, petroleum would not compare very badly with it if bought in a Black Sea port. Considerable interest under the circumstances attaches to an experiment now being made by Mr. A. Suart, of London. He has fitted a large tank steamer, the *Baku Standard*, with appliances for burning astatki, and she recently completed a voyage to Philadelphia from Shields, burning only liquid fuel. She then made a return trip to this country, and has since sailed again. So far as we are aware, this is the first instance on record of a steamship fitted to burn liquid fuel and making an ocean voyage. Hitherto there has always been a breakdown of some kind during the trial. The *Baku Standard* registers 3,705 gross tons. She is 330 feet by 43 feet by 23 feet deep. She carries 1,200,000 gallons of petroleum in bulk, and goes to Russia as well as the United States for oil.

When we come to consider particulars it will be seen that the *Baku Standard* made her voyage under difficulties; instead of the usual crew of firemen and trimmers only three boiler-tenders were shipped. One met with an accident soon after the ship sailed, breaking a collar bone, and was laid up. The result was that the other two men had to stand six hours watch on the rest of the voyage, which they could scarcely have managed if they had had to work hard shoveling coal and cleaning fires. As a fact, however, the fires seem to have needed little or no attention, even in the worst weather, and so far the venture has been quite successful. The consumption of oil was about 20 tons in the twenty-four hours. Taking the oil as being twice as good as coal, this means 40 tons of coal per day, and with good triple-

expansion engines this should give 2,500 indicated horse power. The engines of the *Buku Standard*, however, do not indicate anything like this, and we shall perhaps overestimate the efficiency of the astatki fuel if we say that 20 tons of it were equal to 30 tons of coal, which would give 1,500 to 1,800 horse-power. No doubt the cost of astatki was greater than that of coal, but against this must be set the greatly reduced cost of the fire-room crews—so many fewer mouths to be fed, and so much less wages to be paid.

So far Mr. Suart seems to be contented with the result in all respects but one, and that is the old trouble that has arisen over and over again when petroleum fuel is used in conjunction with surface-condensing engines. A very serious quantity of steam is used in "spraying" the astatki. This represents so much waste, but that is by no means the worst of the matter. The steam thus drawn off from the boiler goes into the chimney in one form or another, and the auxiliary feed has to be employed to make up the loss. Salt water cannot be pumped into the boilers, fresh water enough cannot be carried, and consequently an expensive and heavy evaporating plant must be fitted and worked with liquid fuel. It is very easy to underestimate this item; when we say, however, that during the first voyage of the *Baku Standard*, the weight of steam required to spray the oil exceeded that of the oil, reaching to over a ton an hour, some idea may be formed of what it represents in the way of loss. Allowing 20 pounds of steam to give a horse power—a very ample allowance for a good triple-expansion engine—we see that a ton of steam per hour means about 112 indicated horse power, and allowing that one pound of astatki will evaporate 15 pounds of water, it means that not less than 3,600 pounds, or over a ton and a half of astatki, had to be burned every twenty-four hours for the purpose of supplying fresh water to the boilers. This represents a very considerable addition to the cost, and it certainly seems strange that considerably over 100 horse power should be required to force the fuel into the furnaces. We understand that a different set of burners is now be-

ing tried, and it is hoped that the auxiliary feed will be reduced from 25 tons to 10 tons per twenty-four hours.

There remains for consideration the difficulty involved in handling the petroleum. In the *Baku Standard*, as at first fitted, the stuff was pumped up from special bunkers into two small tanks in the funnel casing, where it becomes more fluid than water, and from these tanks it ran down to the furnaces through suitable pipes. It appears that this arrangement has had to be modified. There is great difficulty in dealing with the oil in the bunkers when they get partially empty. It is well understood that oil-tanks should always be kept quite full. There is absolutely no danger of an explosion when air is kept out; the difficulty is to prevent the accumulation of an explosive mixture of air and petroleum vapor in the bunkers when the level of the oil falls in them. It is very easy to say that means of ventilation can be provided; not so easy to do it. The bunkers in the *Baku Standard* are insulated from the boilers by coffer dams filled with water; but the two service tanks cannot be insulated so, and if they leaked, or any of the pipes or fittings, there would be a serious risk of fire incurred. We believe that two other vessels belonging to Mr. Suart are to be altered to burn liquid fuel; so that the experiment, it will be seen, is being carried out on a scale commensurate with its importance. "Lloyd's," it is worth while to add, are throwing no obstacles in the way; on the contrary, they, as usual, treat the whole subject broadly. The surveyors satisfy themselves that the arrangements for carrying the oil are satisfactory, and then they enter in the register book, "Burning Liquid Fuel, Experimental."

What the outcome of the experiment may be it is impossible to forecast with any pretense to accuracy. It may be that petroleum steamers, as a class, will find it to their advantage to burn petroleum instead of coal. It would seem, however, that the economy must be indirect rather than direct. The fuel, as fuel, will be more expensive than coal, but the handling of it will be much less costly. There may also be a saving in the upkeep of furnaces and boilers. It seems that the introduction of some other method of burning the oil, instead of a steam jet, would be

advantageous. The loss of feed water would thus be avoided. It might be practicable to employ compressed air instead of steam, and it is scarcely probable that 110 horse power would be expended in compressing air in such a ship as the *Baku Standard.* It remains, indeed, to be seen whether mechanical means may not be found for accomplishing the same end. It is a curious fact that, notwithstanding its high inflammability, it is very difficult to burn petroleum at once thoroughly and rapidly. Thus we find that unburned oil may be seen sputtering on the white-hot fire brick opposite the nozzle and then igniting, the fine drops having had to pass through a zone of intensely-heated flame before they could reach the fire tile. Up to the present it is true that nothing has been found to answer so well as the steam jet, but it does not at all follow that finality in this direction has been reached, and there is so little to be said in favor of the steam sprayer in use at sea, that we may expect inventors to work hard to devise something better. Yet when all that is possible has been done the use of petroleum must remain limited, the whole weight of it available for fuel being as nothing compared to that of the coal burned each year.

The superintending engineer of the firm that built the engines of the *Baku Standard,* in a letter to "The Engineer," says that the I.H.P. with coal is from 1,500 to 1,800, and the consumption of coal of average quality about 30 tons a day; that no change has been made in the supply tanks and pipes, except to put cocks below the tanks in the pipes leading to the burners; and that there is nothing in the furnace except a fire brick arch about 2 or 3 feet from the front.

SHIPS.

UNITED STATES.

Indiana.—On the 27th of March this battleship, building by The William Camp & Sons Ship and Engine Building Company was given a builders' trial, off the capes of the Delaware, preparatory to the official trial. During the course of the trial, she made three double runs over a course on Five Fathom Bank 9.65 knots in length, with the following mean results:

	Natural draft.	½ inch.	1 inch.
Air pressure			
Revolutions	110.2	120.5	124.4
Speed	14.08	14.93	15.35

During the first run the draft was assisted by the blowers, but the fire rooms were not closed. The coal used was of inferior quality, and although the ship was not weighted to her trial displacement, the run was made in shoal water, so that there is no doubt that she will more than make her contract speed of 15 knots.

Texas.—The dock trial of the engines of this ship was completed on the 14th of May, with satisfactory results, but too late for insertion in this issue of the JOURNAL.

Ericsson.—This torpedo boat, which has been so long building, was launched at Dubuque, Iowa, on the 12th of May, and will be prepared as rapidly as possible for the trip from Dubuque to Pensacola, where she will be finally completed and tried.

AUSTRIA.

Tegethoff.—This broadside battleship, which was launched in 1878, has recently had entirely new machinery fitted, a change made in her rig and in her battery, amounting altogether to a complete remodeling of the ship. "Mittheilungen aus dem Gebiete des Seewesens," from which this information is taken, says that she was originally a barkentine-rigged, single-screw ship, spreading 14,500 square feet of canvas. She had horizontal, simple back-acting engines, with two cylinders 125 inches diam-

eter and 50 inches stroke, and nine box boilers working under a pressure of 28 pounds. Her speed on the measured mile, with 65 revolutions and 5,864 I.H.P., was 14.37 knots; that on her endurance trial was 13.3 knots, with 60 revolutions and 5,054 I.H.P.

As reconstructed, she is a twin-screw ship with two military masts, each with two fighting tops, the foremast having a search-light platform at one-third its height above the deck, two light-signal yards and an electric signal apparatus.

The new machinery, constructed by Schichau, the well-known torpedo-boat builder, consists of vertical triple-expansion engines, with cylinders 33.86, 55.1 and 84.6 inches diameter by 39.37 inches stroke, each of which is in a separate water-tight compartment. The valve gear is of the Joy type. The air pumps, driven from the H.P. crossheads, are each 17.32 inches diameter and 18.9 inches stroke. The main condensers have a combined cooling surface of 16,114 square feet. There is also an auxiliary condenser. The screws are 14.76 feet diameter and 14.6 feet pitch, measured at 0.7 radius.

Steam is furnished by eight single-ended cylindrical boilers, in four water-tight compartments, working at a pressure of 156 pounds. They have a total grate and heating surface of 538 and 19,978 square feet respectively. There are, besides, two small auxiliary boilers for supplying the auxiliary engines.

In the change from single to twin screws the hole in the stern post was stopped with an ogival-headed plug, and the one at the inboard end of the stern tube was also plugged, and the space between the stern post and the balance rudder filled in with steel plates, which in effect formed a continuation of the stern post, and materially reduced its thickness.

The official trials took place in November of last year. During the six-hours' endurance trial, with natural draft, her mean speed was 15.2 knots with 118.4 revolutions and 7,340 I.H.P. The two-hours' forced-draft trial followed, when, with an air pressure of ½ inch of water, she made 15.5 knots with 125.35 revolutions and 8,950 I.H.P. This power was obtained on a weight of machinery of 225 pounds per I.H.P., while the weight of the original machinery was 432 pounds.

Her new armament comprises six 24-cm. and five 15 cm., 35-caliber Krupp guns, two 7-cm. B.L.R., fifteen 47-mm. rapid-fire, and two 8-mm. machine guns, and two torpedo tubes.

During reconstruction her water-tight subdivision was improved, and a complete drainage system put in.

She is 286.9 feet long, 71.1 feet beam, 24.8 feet draught, and has a displacement of 7,390 tons.

Satellit.—This torpedo cruiser, built by Schichau for the Austrian government, and referred to on page 394 of the discussion on water-tube boilers, is 220 feet 6 inches long, 26 feet 10 inches beam, and has a displacement of about ~~800~~ 540 tons. She has twin-screw triple-expansion engines, to which steam is supplied by four locomotive boilers fitted with Schichau's ash pit forced draft. The grate and heating surface, as given on page 394, are respectively 185 and 8,417 square feet.

Her contract speed was 21 knots, but on her official trial, which was of three hours duration, she made, under unfavorable circumstances, 21.87 knots on 4,792 I.H.P.

The detailed weights of her machinery, as given by the "Zeitschrift des Vereines Deutscher Ingenieure," are as follows:

	Tons.
4 locomotive boilers, including water in boilers, forced-draft fittings, smoke pipe, fittings in fire rooms, and boiler covering	94.57
2 three-cylinder, triple-expansion engines, with surface condensers, evaporators, separators, and all pumps	41.73
Piping, outboard valves, and water in condensers and pipes	6.70
Shaft bearings, stern tubes, propellers, etc	13.06
Circulating pump, fan engines, steam and hand pumps	2.91
Floor plates, ladders, tool chest, work bench, spare parts, stores and tools	5.88
Total	164.85

ENGLAND.

New Battleships.—The British Admiralty has decided to build seven new battleships of the *Majestic* type; five of them will be built in dockyards and two by contract. Of the dockyard ships, the *Cæsar* and the *Prince George* will be built at Portsmouth; the *Illustrious* and the *Victorious* at Chatham; and the *Hannibal* at Pembroke. Machinery for one of them has been ordered from Humphrys, Tennant & Co., for another from Harland &

Wolff, and for a third from Hawthorne, Leslie & Co. The *Jupiter* will be built and engined by J. & G. Thomson, Clydebank, and the *Mars* by Laird Brothers, Birkenhead, each of which firms constructed one of the vessels of the *Royal Sovereign* class.

The new vessels have been designed by Mr. W. H. White, and are 390 feet long, 75 feet beam, and have a load draught of 27 feet 6 inches, with 900 tons of coal in the bunkers, when the displacement is 14,900 tons. The weight of hull is 10,180 tons, and the broadside will be covered from under the water line to the top deck with Harveyized plates 9 inches thick, the depth extending to about 15 feet. The armament consists of four 12-inch breech-loading guns, mounted in pairs *en barbette*, and armored with 14-inch Harveyized plates, and in addition there will be twelve 6-inch and sixteen 12-pounders. The machinery is designed to develop, under natural draft, 10,000 indicated horse-power, giving a speed of 16.5 knots, and under forced draft 12,000 indicated horse-power, giving a speed of 17.5 knots.

Hornet.—A sister vessel to the *Havock*, described in the last number of the JOURNAL, and built also by Yarrow & Co., the only difference in them being that due to the substitution of the Yarrow water-tube for the locomotive boiler. She is 180 feet long, 18 feet beam, and on a mean draught of 5 feet displaces 220 tons. Her engines are 18, 26 and 39½ inches diameter by 18 inches stroke, and are in a common tight-water compartment. There are eight boilers arranged in pairs in two water-tight compartments, each pair facing the other pair in the same compartment, and each pair having its own smoke pipe. The tubes connecting the upper and the lower drums are of copper, 1 inch in diameter, and the downcast tubes at the ends have been omitted. The grates are 6½ feet long, the total grate and heating surface being respectively 165 and 8,216 square feet. The weight of each boiler, with water and fittings, is 5.35 tons, and the saving in weight, over that of the *Havock*, due to the adoption of water-tube boilers is eleven tons. The propellers are three-bladed, 6 feet 4 inches diameter.

The first preliminary trial was made after one-half the boilers had been put on board, and in this condition she made 23.3

knots over the measured mile. A second trial was made after all the boilers were in place, and the mean of six runs over the measured mile gave a speed of 28.02 knots, with 172 pounds steam pressure and 1¼ inches air pressure. The official trial took place on the 19th of March, under favorable conditions, and is thus reported in "Engineering":

The three-hours' trial commenced at the Chapman Lighthouse and ended below the Sunk Lightship. The average speed for the whole time was found to be 27.628 knots per hour.

After this circles at full speed were turned to starboard and port, and generally all the usual tests of machinery and ship were made, all of which were found to be perfectly satisfactory. At full speed and at slower speeds practically no vibration was felt. There was no heating of any parts of the engines, and the boilers made ample steam with a mean air pressure of 1.5 inches. The Admiralty authorities expressed themselves highly pleased with the result in every respect.

A comparison between the results of the *Havock* and the *Hornet* shows that in the former vessel on the official trial approximately 3,500 horse power was indicated, while in the case of the *Hornet* 4,000 horse power, which approximately, is in direct proportion to the cubes of the speeds of the two boats, the consumption of fuel also being in direct proportion to the powers indicated. The air pressure in the stokeholds of the *Havock* was about 3 inches, and in the *Hornet* one and one-half inches.

The runs on the mile resulted as follows:

Steam in boilers.	Vacuum.	Air pressure.	Revolutions.		Speed.	Means.	Second means.	
			Starboard.	Port.				
170 lbs. mean.	26 ins.	Averaged 1¼ ins.	395.9	395.4	26.163	27.322	27.370	27.313
			384.4	386.8	28.481	27.418		
			384.7	392.2	26.355	27.373	27.395	
			391.3	392.7	28.391	27.239	27.306	
			380.0	381.7	26 087	27.128	27.183	
			394.3	394.3	28.169			

Harrier.—A first-class torpedo gunboat, was launched at Devonport on the 20th of February. She is a further development of the *Sharpshooter* type, her dimensions being:

Length, feet,	250
Breadth, feet and inches,	30–6
Draught, forward, feet and inches,	8–9
aft, feet and inches,	9–3
mean, feet,	9
Displacement, tons,	1,070
Weight of hull, tons,	555
Bunker capacity, tons,	100

The engines are twin-screw, vertical, triple-expansion, with cylinders, 22, 34 and 51 inches diameter, by 21 inches stroke, designed for 3,500 I.H.P. under forced, and 2,500 under natural draft, the corresponding speed expected being 19 and 17 to 17.5 knots. The boilers are four in number, of the wet bottom locomotive type, each with two furnaces, and are to work under a pressure of 150 pounds; they contain 169 square feet of grate and 6,106 square feet of heating surface. The screws are 8 feet 3 inches diameter, and 9 feet 4 inches pitch.

Her armament, which will cost $27,000, consists of two 4.7-inch and four 6-pounder rapid-fire guns, one bow torpedo tube, and two broadside tubes. The estimated cost of her construction is $382,000.

Two similar vessels, the *Halcyon* and *Hussar*, are also building at Devonport, and others of the same class by contract.

Hazard.—A vessel of the *Harrier* type, built at Pembroke dockyard, and engined by the Fairfield Co., has completed her natural-draft trials, with the following results:

Steam pressure,	137
Air pressure,	0.82
Vacuum,	24.8
Revolutions,	228.4
I.H.P.,	2 [illegible]
Speed, by patent log,	[illegible].1

The three-hours forced-draft trial gave 3,714 I.H.P., and a little more than 19 knots.

An examination of her boilers after this trial disclosed so many leaks that another trial has been ordered. A like condition of things was found in the *Dryad*, another of the same class.

Speedy.—This torpedo gunboat, fitted with the Thornycroft boiler, was given a trial under natural draft, on the 1st of March, when, with a steam pressure of 177 pounds, she made a speed of 17 knots on 3,071 I.H.P.

FRANCE.

Charlemagne, Saint Louis, Henri IV.—Three battle-ships to be commenced this year.

Their principal dimensions, as given in "Le Yacht," are:

Length, feet,	385.5
Beam, feet,	67.26
Draught, forward, feet,	24.27
aft, feet,	27.56
mean, feet,	25.92
Displacement, tons,	10,951

There are two armored decks—one of 3.5 inches arranged as in the usual type of protected cruisers and armored vessels, worked at the upper water line; the other 1.5 inches thick and worked at the lower water line. The latter is essentially a splinter deck, and its presence is due to the introduction of high explosive shell. Below this second deck are the vitals of the ship—engines, boilers, turret-turning machinery, magazines, etc. The space between the two armored decks is, properly speaking, an enormous coffer dam, and is minutely subdivided. The outboard compartments of it are intended to remain empty, or to be filled with coal in bags. Those in the center contain the auxiliary engines, dynamos, ventilating engines, and ammunition hoisting engines, whose momentary stoppage or even destruction does not endanger the life of the ship.

The armor belt is 6½ feet high, and extends completely around the ship. Its upper edge is 20 inches above the water line, and

its thickness is 15¾ inches at the center, with a height of 27½ inches above the water line. It decreases to the lower part, where it is only 10 inches thick. At the ends the thickness is slightly reduced. Protection above the water line is furnished by a 3-inch belt of armor.

The armament consists of eight 14-cm. guns on the upper deck, and four 30-cm. in barbette turrets. The conning tower has armor 4 inches thick.

There are two military masts, each with two fighting tops.

The motive apparatus consists of three sets of triple-expansion engines, of four cylinders each, operating three screws. Each engine is in a separate water-tight compartment, the axes of the three shafts in the same horizontal plane, and the side screws well forward of the center one.

Steam is furnished by twenty Belleville boilers placed in four water-tight compartments, arranged back to back along the central longitudinal bulkhead, there being one smoke-pipe for each pair of opposite fire rooms.

The power is 14,000 I.H.P. with forced draft, under which conditions the speed is expected to be 18 knots; with natural draft the speed expected is 17 knots.

She carries 680 tons of coal at the normal draft, but can stow about 1,100.

The armament comprises four 30-cm. guns in two turrets, one forward and the other aft; ten 16-cm. R.F.; six 10-cm. R.F.; sixteen 47-mm.; eighteen 37-mm.; four submerged torpedo tubes, two forward and two aft, firing on the sides, and six above-water tubes. The forward turret is 27.88 feet above the water line, and the after one 21.32 feet.

The fire rooms being fore and aft and outboard the boilers, the coal bunkers and magazines are outboard and in line with the bunkers, and therefore in front of the boilers.

The boilers occupy about 125 feet in the length of the ship, and, with fire rooms, about 36 feet in width. The engine rooms, which are opposite each other, are about 50 feet in length.

Suchet.—A second class cruiser, of 3,374 tons displacement, was recently tried with the following results:

	Knots.	*I.H.P.*
March 20, 2 hours, forced draft, .	20.417	8,842
April 9, 6 hours, assisted draft, .	18.238	6,210
April 16–17, 24 hours, natural draft, .	16.73	4,119

The revolutions on the last trial were 105.

The *Suchet* is 311.6 feet over all, 288.7 on water line, 40.4 beam, 19.5 mean draught, has a displacement of 3,374 tons, and was designed for a speed of 20 knots on 9,000 I.H.P. She has horizontal, triple-expansion engines, with cylinders 32.3, 47.25 and 75.6 inches diameter by 39.37 inches stroke.

Forban.—Sea-going torpedo boat building by Normand, designed for a speed of 30 knots. She is 144 feet 3 inches long, 15 feet 3 inches beam, 10 feet extreme draught, and has a displacement of 130 tons. She will have twin-screw, triple-expansion engines of 3,200 I.H.P., and will be fitted with the Normand boiler.

JAPAN.

Tatsuta.—A torpedo cruiser building by Armstrong, Mitchell and Co. for the Japanese government, was launched on the 6th of April. She is 240 feet long, 27½ feet beam, and has a displacement of 875 tons. The speed expected is 22 knots with 5,500 I.H.P. Her armament comprises two 4.7-inch and four 3-pounder rapid-fire guns and five torpedo tubes.

RUSSIA.

Paris.—A new battleship of 8,800 tons displacement is to be laid down at Nicolaieff. She will be 343.74 feet long and 67 feet beam, with engines of 8,000 I.H.P., which are expected to give her a speed of 16.5 knots. She will have an armor belt 15¾ inches thick for four-fifths the water line, and the remainder 11.8 inches. Above the belt there will be 5-inch armor for the protection of the secondary battery. The armament will be two 30-cm. guns in the forward turret, and two in the after one, six 15-cm. rapid-fire guns and twenty of small caliber, besides six torpedo tubes. She will have sixteen single-ended cylindrical boilers in four water-tight compartments, and will stow 800 tons of coal.

MERCHANT STEAMERS.

Jamestown and *Yorktown.*—These two vessels, which are the latest additions to the Old Dominion Line, were designed and built by the well-known Delaware River Iron Shipbuilding and Engine Works, Chester, Pa. They are of steel, 300 feet on the water line, 322 over all, 40 feet beam, 26¼ feet depth of hold, and are of 2,126 tons, net. The greatest draught is 20 feet.

The engines are triple expansion, with cylinders 28, 44½ and 73 inches diameter, by 54 inches stroke, and were designed for 3,000 horse-power. There are four boilers 13 feet 9 inches diameter, and 12 feet 6 inches long, each with three 44-inch corrugated furnaces, and designed for a working pressure of 180 pounds per square inch. The total grate surface is 300 square feet. The smoke-pipe is elliptical in section, 8 feet 6 inches by 6 feet 10 inches, and the top of it is 86 feet above the grates.

The screw propeller is four bladed and adjustable, of manganese bronze, 15 feet 6 inches diameter, and 21 feet 6 inches pitch.

These vessels have been fitted with the latest improvements, and are said to be models of comfort and safety. They make the run from New York to Norfolk in 18 to 19 hours. The *Jamestown*, on one of her first trips, made the run at an average speed of 15.75 knots per hour.

Northwest.—This vessel recently made a run of 120 miles to test her machinery preparatory to the official trial, and on the latter part of it worked up to 108 revolutions, with a pressure of 200 pounds in the boilers and 175 at the engines. During the greater part of the run 27 of the 28 Belleville boilers were in use. Everything is reported to have worked satisfactorily.

Unique.—This is the name which will be given to a small

steamer building for Mr. McElroy, of St. Clair, Mich., and intended for passenger service between Detroit and Port Huron. The hull is of wood, built on the St. Clair river, and is 175 feet long, 24 feet beam over guards, 11 feet depth of hold amidships.

The engine, building by the Frontier Iron Works, Detroit, is of the quadruple-expansion type, with cylinders 15, 22½, 30 and 42 inches diameter by 18 inches stroke, designed for about 1,500 I.H.P. at 300 revolutions. The H.P. cylinder is forward, and the others follow in regular order; the H.P. and first I.P. cranks are opposite each other, as are also the second I.P. and the L.P., which are 90 degrees with the former. The crank shaft is hollow, 7¼ inches diameter with journals 10 inches long. The pins are of steel, hollow, and the cranks of cast steel with counter balances cast on. The propeller is of bronze, left handed, four bladed, and 7 feet diameter.

There will be two Ward torpedo-boat boilers, each with 50 square feet of grate and 2,000 of heating surface, designed for a working pressure of 225 pounds per square inch. They will be placed fore and aft, each with its own smoke-pipe, and will be fitted for working under forced draft. In this boiler, there is an upper steam and water drum, and two lower water drums connected with the upper one by bent tubes 1½ inches diameter. The feed is introduced into a pan in the upper drum, and the pan so arranged that the circulation is down the outer rows of short tubes. Besides the short tubes, which go to the lower part of the drum, there are longer ones which discharge into the upper part of it. The weight of both boilers in steaming condition is 18 tons.

The name *Orinda* signifies "matchless" or "incomparable," and the "Marine Review," from which the greater part of this description is taken, says that the average running speed will be 20 miles an hour, though other papers put it as high as 25. Name since changed to *Unique*.

Marie Henriette.—This paddle steamer, intended for the Ostend-Dover route, whose trial was recorded on page 1086 of volume V of the JOURNAL, has recently been fully described and

illustrated in "Engineering," and in the "Zeitschrift des Vereines Deutscher Ingenieure," from which the account here given is made up. She was built by the Société Cockerill, of Seraing, Belgium, in competition with the *Leopold II*, which was built by Messrs. Denny, of Dumbarton, Scotland, the stipulated speed in each case being 21½ knots, with a premium of $3,750 for each tenth of a knot above the contract speed, and a penalty of $2,500 for each tenth of a knot below. If the vessel failed to make 21 knots, she would be rejected. The limit of draught was 9 feet 3 inches, and on this she was to carry 70 tons of coal and 20 of baggage. Both vessels were tried in the Firth of Clyde, the trial consisting of four runs over a base 13.666 knots in length, between the Cloch and Cumbræ lights, and resulting in a speed of 21.995 knots for the *Leopold II*, and 22.2 for the *Marie Henriette*, the mean draught of which, with the prescribed weights on board, was 9 feet. The I.H.P. on this trial was 8,200, the revolutions 53, and the steam pressure 120 pounds.

The *Marie Henriette* is of the following dimensions:

Length between perpendiculars, feet, . .	340
Breadth, extreme, of hull, feet, ...	38
over guards, feet and inches, . .	76–7
Depth from top of keel to main deck at side, feet,	15
to promenade deck, feet and inches, . .	22–9
Displacement at mean draught of 9 feet, tons, .	1,525

The propelling engines are of the compound type, with two cylinders, inclined, but parallel to each other. The cylinders are placed aft of the paddle shaft. The seats of the cylinder are parallel to the axis of the cylinder, and not to the keel of the ship, and the triangular space which is thus left free is filled by a steel casting bolted to the foundation, which is horizontal. The framing of the engine is in four parts, of cast steel, bolted to the keelson of the ship, the upper part, supporting the crank-shaft bearings, being connected to the cylinders by the columns forming the crosshead guides. The frames are braced together trans-

versely. The high-pressure cylinder has a diameter of 60 inches, and the low-pressure of 108 inches, the stroke being 84 inches. The cylinders have certain peculiarities which are worthy of note. The tops are cast in one piece with the body, but they have at their middle an opening with a cover, having the stuffing box of the piston rod. This opening is made so large that after drawing up the cover on the rod there remains sufficient space around the rod to enable a man to enter the cylinder. This dispenses with a separate manhole, and facilitates the casting of the cylinder. There is a relief valve $5\frac{1}{8}$ inches in diameter, loaded with a spring on the top, and another in the middle of the cylinder cover placed on the manhole. The pistons are of cast steel, and are conical in form. They have a simple cast-iron ring, $1\frac{3}{16}$ inches thick and 5.9 wide, behind which are flat springs of the ordinary type. On the lower portion there are no springs, the width of the piston being supported by the aid of a packing piece, while the tongue of the ring is on the top.

The piston rods are of forged steel, $10\frac{1}{4}$ inches in diameter, and pass through stuffing boxes with metallic packing. The crossheads, of forged steel, have two journals to receive the brasses of the connecting rods, which are also forged. Each of the two ends of the crosshead is bolted to a U-shaped piece of forged steel. The branches of the U are above and below the guide bars, and are supported by set bolts on the slippers, which latter are of cast iron lined with white metal. This arrangement insures that the slippers are pressed only on the center, and are not subject to the slight flexure of the crosshead. They are thus supported very uniformly on the guides, and the guides have shown, after a small time of working, a surface equally polished on both their length and breadth. The crosshead pins are 12.6 inches diameter and 11.69 inches long. The cylinders are connected to the bearings of the crank shaft by forged steel columns forming the crosshead guides. The portion of these columns on which the shoes run is I-shaped; the other parts are round. The main connecting rods are 14 feet long, and of forged steel. The brasses of the fork embracing the crosshead are of gun

metal. Those of the crank are of cast steel lined with white metal, the lower part being round, so that they can be taken out without removing the shaft. The covers are also of forged steel. The crank shaft is of forged steel built up, and hollow in the body and in the crank pins. It has a uniform diameter of 19 11/16 inches. The main bearings are 29½ inches long, and the crank pins have a length of 23⅝ inches. The diameter of the hole in the end sections of the crank shaft is 7.28 inches, reduced to 5.91 at the crank webs; in the section connecting the adjacent cranks of the two cylinders it is 11.81, and reduced as before to 5.91 in the crank web; in the crank pins it is 11.42. The hub of the crank is 37.4 inches diameter, 11.81 inches thick, varies in width from 11.81 to 23.62 inches, and is 72.4 inches in length over all. The length of crank shaft over all is 17 feet 10.56 inches. The cranks are at 90°, and each pair and its pin is in one piece, so that the crank shaft is in five pieces. The crank shaft carries on each extremity a flange, 35.43 inches diameter and 4.92 inches thick, for coupling to the paddle-wheel shaft.

The paddle-wheel shafts are of forged steel, hollow, and of the same diameter as the crank shaft. The external supports of the paddle shafts are of cast steel. They have only a single brass below. The cover is hollow, and without a brass. The brass is of cast steel lined with white metal, and is 4 feet 3 3/16 inches long.

The steam distribution is made to the high-pressure cylinder by a cylindrical piston valve which travels in a hard cast-iron bushing. The low-pressure cylinder has two flat double-ported valves which work on hard cast-iron seats. The two valves are placed together and work together. With two valves there are smaller faces, and there is therefore the minimum of deformation under the influence of heat. The valve motion is obtained from a single eccentric, the link oscillating on a fixed center. The link does not act directly on the valve, but on a lever fixed to a movable center, one extremity of which is in connection with the crosshead, by which is obtained the necessary lead. This arrangement is known as the Heusinger or Walschaert valve

gear. The oscillating links are fixed on a rigid cross beam of cast steel, connecting the top of the four frames of the engine. The linking up is effected by steam and hydraulic reversing gear. A simple hand force pump serves to change the gear when there is no steam.

The condenser is cylindrical, made of steel plates and brass ends, and is placed athwartship beneath the L.P. guides. The tube plates, which are of brass, are ¾-inch thick, and the tubes ¾-inch outside diameter. The cooling surface is 10,226 square feet.

There are two vertical single-acting air pumps, each 37.76 inches diameter and 25.51 inches stroke. They are worked by bell cranks from the crossheads. All parts of the pumps, except the cover, are of bronze; the cover is of cast iron; the valves of rubber.

The paddle wheels are 22 feet 4 inches diameter, and each have nine curved feathering buckets 4 feet 4 inches deep by 15 feet wide. The wheel frames are of wrought iron, and are secured to cast-steel centers. The buckets are of plate steel, and are strengthened by doubling where the bending and twisting are greatest. All joints run brass on brass except the eccentric, where it is brass on lignum vitæ. The pin in the center of the eccentric is fixed in a steel casting secured to the guard by bolts. The port shaft carries a small toothed wheel, by means of which the main engines can be moved by a small engine. They can also be turned by manual power.

The condensing water is supplied by two centrifugal pumps, one on each side, with runners 34 inches diameter. They are driven by 10 by 10-inch vertical single-cylinder engines.

The boilers are fed by two Weir pumps, 11½ by 16½ by 20 inches stroke, each arranged to work independently or in conjunction. There is another Weir pump, 8 by 9 by 18, fitted for fire and bilge and general auxiliary purposes.

The feed pipes are in duplicate, and either pump can supply water to both or any of the feed pipes. These pumps are arranged so that they can circulate water in the boilers, and also

empty them. They can also take water directly from the condensers. A Weir's distiller is fixed in the engine-room to make up the feed. This distiller is 3 feet 6 inches in diameter, and 3 feet 9 inches long, and can supply 12 tons of distilled water a day.

Steam is furnished by eight single-ended return-tubular boilers, working under 115 pounds pressure, four forward and four abaft the engines, arranged in four fire rooms, and for working under forced draft with closed fire rooms. They are of steel, 13 feet 4 inches diameter and 10 feet 3 inches long, each with three corrugated furnaces 37¼ inches inside diameter. The furnaces are so made that they may be removed, and the joint at tube-sheet end is protected by a ring of fire brick. Each furnace has a separate combustion chamber. The grate surface is 465, and the heating surface 14,100 square feet.

The ordinary tubes are of steel, 2½ inches diameter and 6 feet 6¾ inches long between tube sheets, and .118-inch thick; the stay tubes 2 5/16 inches diameter and .177 to .216-inch thick. Neither the ordinary nor the stay tubes are beaded over at the ends, but simply expanded.

The screw stay bolts are spaced 7⅞ inches apart, screwed into both sheets, with nuts on the fire side.

There are two elliptical smoke-pipes.

For forced draft, there are four 60-inch double inlet fans, each run by a Chandler motor. They take air from the upper part of the engine room, and each pair delivers into a rectangular duct which goes around the boilers, and has an opening in front of each with a slide by means of which the air can be regulated.

Priscilla.—This magnificent side-wheel steamer, described on page 848 of Volume IV, under the name of *Mayflower*, has just been completed for the Fall River line by the W. & A. Fletcher Co. of Hoboken. For the benefit of those members who may not have at hand the number referred to, the following particulars are repeated from the "Marine Journal."

Length on load water line, 423 feet 6 inches; length over all, 440 feet 6 inches; beam, 52 feet 6 inches; beam over guards, 93

feet; depth moulded at lowest point of sheer, 20 feet 6 inches; mean draft, loaded, 13 feet; displacement, loaded, 5,030 tons; mean draft, light, 12 feet; displacement, light, 4,550 tons.

The main engine is of the double, inclined compound surface-condensing type, of 8,500 horse power. There are two high-pressure cylinders, each 51 inches diameter, side by side, forward of the main shaft, and two low-pressure cylinders, each 95 inches diameter, side by side, abaft the main shaft, all having a stroke of 11 feet.

The main engine shaft is in three sections, the two outboard ones having the driving, and the center section the drag cranks, there being two sets of cranks set at right angles. Each crank pin is connected to the connecting rods of a high and a low pressure cylinder. Owing to the inclination of the cylinders, the connections of but one cylinder can be thrown upon a dead center at the same time, thus insuring a very steady and even rotation of the main shafts, an equal distribution of power, and minimum vibration. Each low-pressure cylinder has its own air pump and surface condenser. The air pumps are vertical and are worked from the low-pressure crossheads. There are two independent centrifugal circulating pumps for each condenser. Total cooling surface of condensers, 16,000 square feet. The main steam chest valves are of the double poppet type, and are worked by double eccentrics and links. The high-pressure cylinders have adjustable drop cut-offs; the low-pressure cylinders fixed "Stevens'" cut-offs.

The paddle wheels are of the feathering type, 35 feet in diameter outside of buckets. There are 13 curved steel buckets in each wheel, each bucket being 5 feet deep by 14 feet wide.

The main boilers, ten in number, are of the single-ended Scotch type, and were built for a maximum steam pressure of 150 pounds. Each boiler is 14 feet mean diameter by 14 feet 6 inches in length, and contains three corrugated furnaces, each 45½ inches diameter, and 184 Serves' patent ribbed tubes 3½ inches diameter. The boilers are fitted both for natural and forced-draft under grates. The forced draft is supplied by two large Sturtevant

fans driven by independent engines. These fans, being placed in the main engine department, serve also to ventilate the engine room. The total grate surface of boilers is 850 square feet. Total heating surface, 35,400 square feet.

An extra large donkey boiler, and two large fire and wrecking pumps are on the main deck. The pumps are so arranged and connected that they may for all purposes work independently or together. There is also a large bilge pump in the engine department below deck, forming in all three complete wrecking and four very complete fire systems extending the full length and throughout the vessel. In addition, there are a number of hand pumps in various parts of the vessel.

The electric plant of the steamer consists of 1,900 incandescent lamps of 16-candle power each, supplied by three multipolar, direct-coupled, compound-wound dynamos, driven by horizontal "Ideal" engines, having cylinders 11 inches diameter by 12 inches stroke. The capacity of each dynamo is 400 amperes and 125 volts.

After a highly successful trial trip on the Hudson and in New York Bay, the *Priscilla* left for Newport, where she will receive her outfit. The run from New York to Newport was made at the rate of 20.2 miles an hour.

Favorite.—A small excursion steamer, built at Tompkins Cove, on the Hudson, is of interest from the fact that she will be a triple-screw steamer. She is 141 feet long, 36 feet beam, 9 feet depth of hold, and will draw about 4 feet of water. Her engines are compound, with cylinders 11 and 23 inches diameter, by 14 inches stroke. Her boilers are two in number, 6 feet diameter, and 14 feet long, designed for a working pressure of 150 pounds.

Campania and *Lucania.*—The following statement of the time of the voyages made by these steamers is from a tabular summary in "Engineering," giving the daily runs, time of voyage, etc.

CAMPANIA.

No. of voyage.	Distance.	Time. d.	h.	m.	Average speed.	Mean speed for round voyage.
1	2,858	6	8	51	18.70	20.00
	2,927	5	17	32	21.29	
2	2,812	5	21	15	19.92	20.275
	2,895	5	20	18	20.63	
3	2,880	5	16	22	21.11	21.01
	2,913	5	19	18	20.91	
4	2,796	5	15	24	20.65	20.495
	2,825	5	18	57	20.34	
5	2,781	6	2	57	18.93	19.885
	2,816	5	15	4	20.84	
6	2,802	5	18	22	20.25	20.16
	2,820	5	20	30	20.07	
7	2,783	5	13	37	20.83	21.00
	2,799	5	12	15	21.17	
8	2,790	5	20	45	19.83	20.215
	2,820	5	16	54	20.60	
9	2,863	6	13	2	18.24	19.70
	2,871	5	15	44	21.16	
	Mean speed for nine round voyages					20.304

LUCANIA.

1	2,751	5	16	19	20.18	20.235
	2,785	5	17	16	20.29	
2	2,782	5	14	1	20.76	20.86
	2,802	5	13	41	20.96	
3	2,780	5	13	2	20.90	19.42
	2,853	6	15	0	17.94	
4	2,785	5	15	52	20.49	20.71
	2,817	5	14	35	20.93	
5	2,876	6	16	17	17.94	19.855
	2,900	5	13	14	21.77	
6	2,907	5	18	47	20.94	21.285
	2,892	5	13	40	21.63	
	Mean speed for six round voyages					20.394

The best time has then been made by the *Campania*, 5 days 12 hours and 15 minutes, although the *Lucania* has made the greatest speed, not only for the mean of the round voyages, but also for a single and a round voyage, as well as for the day's run in each direction, her biggest runs being 546 miles when running to the westward, and 516 to the eastward.

YACHTS.

STEAM YACHT RACE.

On the 27th of February, there was a steam yacht race in the Mediterranean, under the auspices of the Nice Yacht Club, from Cape Martin to Genoa, a distance of 75 nautical miles. The contesting yachts were the *Fauvette*, owned by Mr. E. A. Perignon, the *Namouna* by Mr. James Gordon Bennett, the *Foros* by Mr. A. Kousnetzoff, the *Roxana* by Prince Leuchtenberg, and the *Eros* by Baron A. de Rothschild. They finished in the following order:

	Time.		*Mean speed.*
	hrs.	*min.*	*knots.*
Fauvette, ...	4	57	15.15
Namouna, ...	5	03	14.8
Foros, ...	5	05	14.75

The *Roxana* and the *Eros* followed a few minutes later, but their time was not taken.

The *Fouvette* is 420 tons yacht measurement, the *Namouna* 740, the *Foros* 1,086, the *Roxana* 812, and the *Eros* 737. The *Fauvette* was designed by Mr. Dixon Kemp, and built last year by Ramage and Ferguson Leith, Scotland. The engines of the *Fauvette*, *Namouna* and *Foros* are of about the same power, 1,100 I.H.P. The *Roxana* is of about 1,000 I.H.P., and is fitted with two Oriolle boilers.

The *Fauvette* is 177 feet long, 22 feet beam, 18 feet depth of hold, and draws about 12 feet of water. The *Namouna* is 219 feet 4 inches long, 26 feet 4 inches beam and 18 feet depth of hold.

This race was followed by a handicap race, the conditions of which were arranged by Mr. James Gordon Bennett, who offered a prize of 5,000 francs to the winning yacht. Four yachts started, but only three finished, the race being won by the

Roxana, in 3 hours 32 minutes 5 seconds, with *Foros* second, and *Eros* third. The *Foros* was giving the *Roxana* 3, and the *Eros* 14 minutes, but the actual time of the *Foros* was nearly 6 minutes more than that of the *Roxana*. The distance was 52 miles, so that the average speed of the winning yacht was 14.7 knots.

Margarita.—Formerly the *Semiramis*, was recently purchased by Mr. Anthony J. Drexel, of Philadelphia. She was designed by Mr. Alfred H. Brown, of London, and built by Ramage & Ferguson, Leith, from whose yard she was launched on the 30th of May, 1889. She is 206 feet over all, 194 feet on the water line, 27 feet beam, 16 feet depth of hold, and of 703 tons.

Her engines are triple-expansion, 18, 29 and 47 inches diameter, by 33 inches stroke, of about 750 I.H.P., working with a boiler pressure of 160 pounds. Her machinery and bunkers occupy about 48 feet in length of the vessel. She was built with an eye to comfort rather than speed, and is said to be a model yacht. She stows about 150 tons of coal.

On her trial in the Firth of Clyde, July 3, 1889, she made 12.5 knots, and is said to be able to make 12 knots at sea.

Formosa.—A two-masted steam yacht building by the Atlantic Works, East Boston, for Mr. Fabyan, of New York, was launched on the 18th of April. She is of steel, 157 feet over all, 132 on the water line, 22 feet beam, and draws 11½ feet. She will have a triple-expansion engine of 850 horse-power, steam for which will be furnished by two Almy water-tube boilers.

Pilgrim.—This sailing yacht is being converted into a steam yacht, and will have twin-screw compound engines of 500 horse-power, and two Almy boilers.

Zeta.—A steel schooner-rigged steam yacht, designed and built by Ramage and Ferguson. She is 133½ feet long, 20½ beam, and 11¾ moulded depth. Her engine is triple-expansion, with cylinders 12, 19 and 30 inches diameter by 21 inches stroke, steam being furnished by a cylindrical boiler working at a pressure of 180 pounds per square inch. Amongst her fittings is a naphtha launch made in this country.

——. —The same firm is building, from the design of Mr. G. L. Watson, an auxiliary three-masted steam yacht, of about 650 tons, for Lord Charles Gordon-Lennox. She is 190 feet long, 29 feet beam, and 18 feet depth of hold. Her engines are triple-expansion, of 700 I.H.P., with cylinders 14, 23 and 36 inches diameter by 27 inches stroke. Steam is furnished by two Belleville boilers. She has a Bevis feathering screw propeller.

Another steam yacht by the same firm is 149 feet long, 20 feet beam, 12 feet depth moulded, and of 280 tons. She has a triple-expansion engine with cylinders 13, 21 and 34 inches diameter by 22 inches stroke, and a cylindrical boiler working at 165 pounds pressure. She is for Colonel Henry Platt, of North Wales.

BOOK REVIEWS.

SEATON AND ROUNDTHWAITE'S POCKET BOOK OF MARINE ENGINEERING RULES AND TABLES. D. Van Nostrand Co., New York. Price, $3.00.

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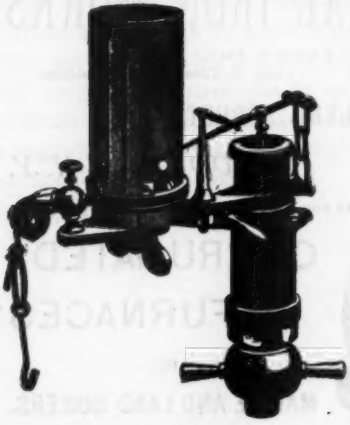

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